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PUBLISHED BY THE SOCIETY OF AUTOMOTIVE ENGINEERS, INC.

D. G. Roos, President — David Beecroft, Treasurer

John A. C. Warner, Secretary and General Manager

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The Society is not responsible for statements or opinions advanced in papers or discussions at its meetings or in articles in the JOURNAL.

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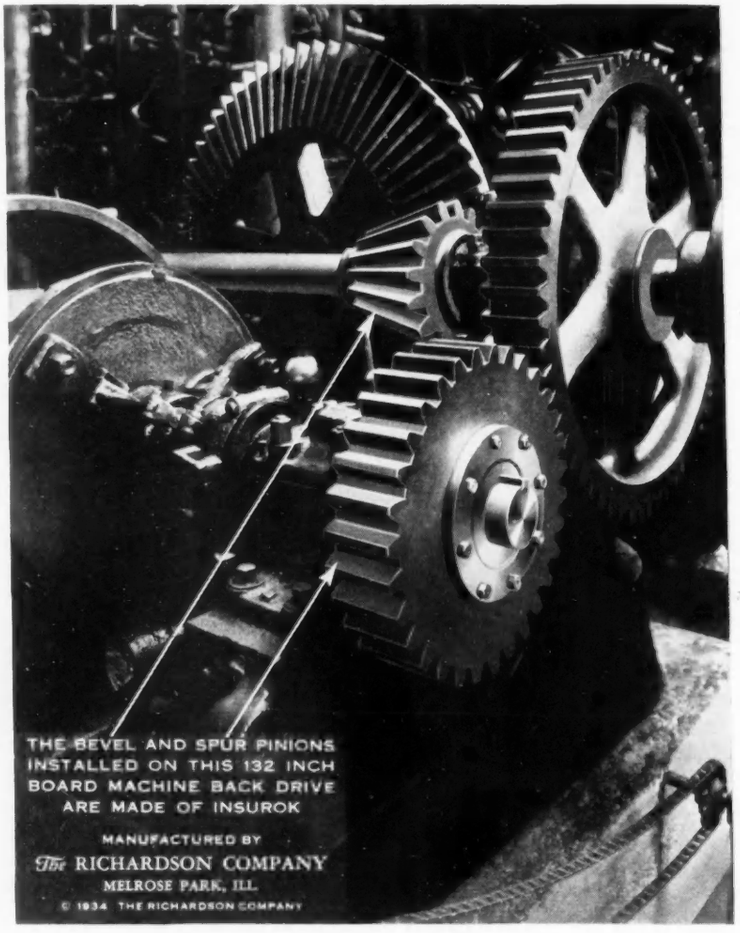
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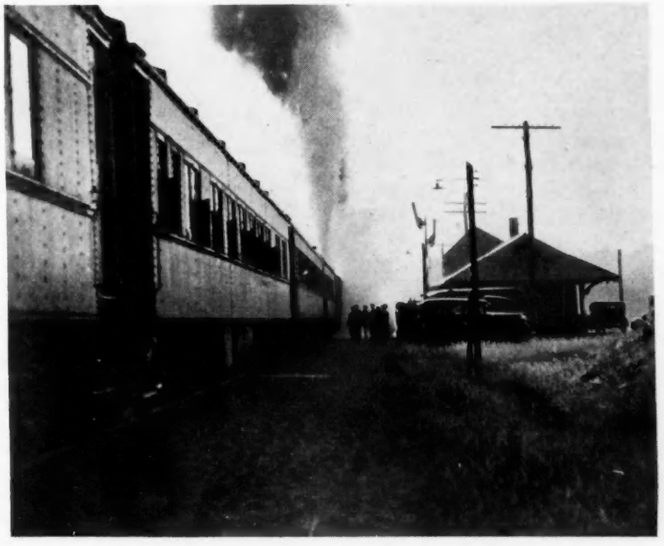
1934 Summer Meeting Best in Years



FUEL and lubricant problems were much in the limelight at the Summer Meeting of the Society of Automotive Engineers held at Saranac Inn, N. Y., June 17 to 22. Attendance was well over the 500 mark and exceeded that of any regular summer meeting since 1929. While fuel and lubricant questions were debated at seven of the 17 technical sessions, streamlining, noise reduction, railcars, accident prevention, airplane propeller and shock-absorber developments, and automobile body designs all came in for detailed exposition and comment.

Twenty-eight technical and administrative committee meetings, a business session, a thrilling adventure talk by Lowell Thomas, and several afternoons of outdoor and indoor sports combined with the formal engineering sessions to fill six days of constructive and pleasant activity. Mr. Thomas, now Sunoco's radio reporter, captivated his audience with stories tall but true and furnished a high spot in the entertainment program.

Digests of all the papers presented appear on pages 21-26 of this issue, while a detailed summary of the discussion at each session will appear in ready-reference form in the August issue. Three summer meeting papers are printed in full on later pages of this issue, while others will be presented in coming months.

Out of the wealth of specialized engineering data and discussion generated by the meeting, however, a number of highlights of general interest stood out for those who scanned the meeting as a whole.

Things became lively at the very start, when, at the business session, opposition was voiced to some of the changes in the Constitution which were presented by J. H. Hunt for the Constitution Committee with the approval of the Council. Particularly sharp exchanges featured the discussion of the proposal to change Section C46, to provide clearly for the rights of alternates for members of the Nominating Committee and to clarify the provisions as regards proxy voting at Nominating Committee meetings, with J. F. Winchester and F. C. Horner opposing the changes and J. H. Hunt, President Roos and others favoring the Council-endorsed proposals.

Since no vote can be taken on such amendments at the time of proposal, only discussion of the measures transpired at this session. Details of the proposed amendments together with full discussion is certain to be brought clearly to the attention of the entire membership before final action takes place many months hence.

Lack of adverse comment indicated general support for perhaps the most important Constitutional change suggested at the business session—a proposal presented by Councilor Walter T. Fishleigh which would constitute all the Past Presidents of the Society into an Advisory Committee whose function it would be to discuss any matters referred to it by the Council or any other problems which in its opinion concern the interests of the Society. This Advisory Committee would be empowered to report to the Council or direct to the membership any recommendations approved by a majority of its full membership.

Immediately following the business session on Monday evening, H. T. Strong, William Wiese & Co., entertained an audience of over 200 with a talk about revelations in the realm of color which was featured by a series of intensely interesting demonstrations illustrating basic principles of color production and combination.

The inefficiency of the human ear as a factory tool for use in directing the steps in noise reduction, despite the fact that it is an admirable instrument for judging the final result, was brought out early in the meeting by Dr. E. J. Abbott. About 75 attended the session at which he spoke. Noise-measuring devices are necessary, he pointed out, but can be

Dr. E. J. Abbott,
Speaker,
Sound Session

E. L. Tirrell,
Speaker,
Motor Vehicle Design
Session

W. C. Keys,
Chairman,
Sound Session

A. L. Clayden,
Chairman,
Passenger Car Fuels
and Lubricants

Arthur Nutt,
Speaker,
Cooling, Detonation
Session

F. L. Faulkner,
Speaker,
Motor Vehicle Design
Session

F. M. Young,
Chairman,
Diesel Design Session

H. D. Hill,
Chairman,
Diesel Fuels Session

S. W. Sparrow,
Speaker,
Bearings, Lubrication
Session

L. V. Newton,
Chairman,
Motor Vehicle Design
Session

C. F. Marvin, Jr.,
Speaker,
Design, Combustion
Session

Louis Schwitzer,
Speaker,
Engine Session

H. C. Mougey,
Speaker,
Car Fuels, Lubricants
Session

K. M. Wise,
Chairman,
Engine Session

A. E. Becker,
Speaker,
Diesel Fuels Session

Lowell H. Brown,
Speaker,
Streamlining Session

C. V. Johnson,
Speaker,
Shock Absorber Session

H. T. Strong,
Speaker,
General Session

F. W. Caldwell,
Speaker,
Propeller Symposium

F. K. Glynn,
Chairman,
Accident Session

H. K. Cummings,
Chairman,
Cooling, Detonation
Session

E. W. Upham,
Speaker,
Car Fuels, Lubricants

W. T. Fishleigh,
Chairman,
Streamlining Session

Robert Insley,
Chairman,
Design, Combustion
Session

Herbert Chase,
Speaker,
Streamlining Session

D. G. Roos,
Chairman,
Business Session

Hans Fischer,
Speaker,
Diesel Design Session

Alex Taub,
Chairman,
General Meetings
Committee

Thomas Barish,
Speaker,
Propeller Symposium

C. M. Larson,
Speaker,
Bearings, Lubrication
Session

Austin M. Wolf,
Speaker,
Truck Design Session

effectively utilized only by skilled users. "Sounds and noises are very complex," he said, "and to work effectively the investigator must have a considerable background of knowledge and experience in noise measurement to obtain the necessary data with a reasonable number of readings and to plan the quieting moves. . . . Meters are important, but the ability and knowledge of the investigator are far more so. If these two are properly combined, successful noise reduction on the basis of sound measurement is a proven fact."

Emphasizing further this same viewpoint, he added that "Noise problems involve such a peculiar combination of small things which are all important, and large things which are negligible that our familiar engineering judgment often leads one far astray, and one must accumulate a rather special set of thumb rules for noise work."

Accident Study Urged

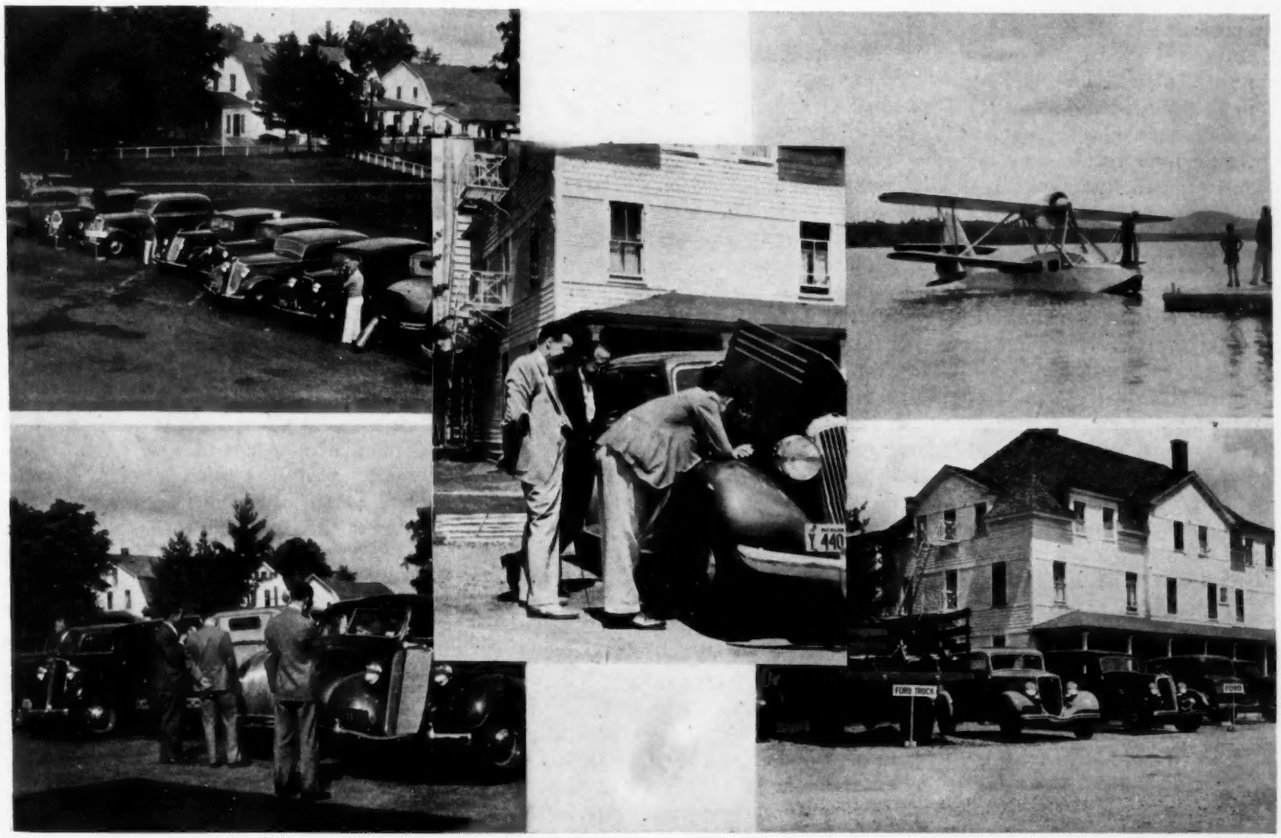
J. M. Orr's paper about accident control in fleet operation, brought from David Beecroft the definite suggestion that the S.A.E. as a Society should get into the accident prevention situation more fully than in the past, a suggestion which gained approval at the same session from M. C. Horine and F. C. Horner. "I am not suggesting that the Society touch on any of the political aspects of this accident situation," Mr. Beecroft explained, "but feel that its participation should take the form of helping to develop some badly needed accurate information on various technical phases of the problem."

With the total economic loss of an accident to a fleet vehicle estimated at $400 to $500 and with some bus companies paying out 5 per cent of their revenues for accident claims and insurance premiums, it is natural that considerable interest should be centered in Mr. Orr's report which he made as chairman of a committee appointed last January by Vice-President Leonard V. Newton of the Transportation and Maintenance Activity.

Representing analysis of composite opinions of leading fleet operators, traffic and safety engineers, state vehicle commissioners and others, the report concludes that: (1) the driver is the most important factor in accident prevention; (2) the the cost of adequate accident prevention activities is negligible as compared to the cost of accidents; (3) that accidents are increasing; and (4) that current prevention activities are not sufficient to stop the upward trend.

Belief in the efficacy of more rigid laws as aids to accident reduction was indicated by Mr. Orr when he said: "The need for driver education, adequate drivers' licensing laws and

Transportation Revue Introduces Cars to Men



More than 300 who attended the Summer Meeting inspected and drove the passenger cars, trucks and the airplane which made up the Transportation Revue, a new feature on Summer Meeting programs.

Buick, Chevrolet, Chrysler, Ford, Graham, Hupmobile, Olds, Pierce-Arrow, Pontiac and Studebaker cars and Ford and Buda-Diesel trucks were shown.

The cars were sent by the factories concerned and the "revue" was very informal. Any one who attended the meeting had an opportunity to drive the cars long enough to get the feel of them and learn about their mechanical features from the engineers or demonstrators who were in charge.

Every one seemed to enjoy the revue and—here's news—an engineer in charge of one of the cars made a sale on the spot.

(Left to Right) J. M. Orr, speaker, Accident Prevention Session; Ferdinand Jehle, chairman, Bearings and Lubrication Session; A. G. Herreshoff, chairman, Truck Design Session; C. O. Guernsey, speaker, Railcar Session; W. D. Teague, speaker, Body Design Session; L. P. Kalb, speaker, Engine Session

uniform traffic regulations is plainly seen. . . . Unfit and unsafe vehicles should be cleared from the roads, probably by compulsory inspection requirements.

M. C. Horine re-emphasized the axiom that "Accidents don't happen; they are caused," urging that accident reporting forms should be so revised as to evince more interest in what caused the accident and less in the happening itself.

F. C. Horner expressed the belief that many accidents occur when cars try to pass trucks which do not stay over on the right of the road and that if the fact were known in each case, one drunken driver probably would be found involved in most accidents involving two cars.

Uniform Truck Rating

In the course of his discussion of how lightness may best be incorporated in truck design, Austin M. Wolf revived the drive for final establishment of a uniform code on truck rating. Despite long and constructive effort along these lines in the past, Mr. Wolf believes still that "the subject requires careful thought and definition to protect the industry against wrong interpretations by the buyer or the legislator."

Asking the question, "How durable should a vehicle be?" Mr. Wolf pointed out that many truck parts still are as good as new when the truck as a whole is ready for the scrap pile. He suggested that parts might be designed for "controllable wear" as well as imperceptible wear.

In discussion of the Wolf paper, J. F. Winchester brought out many interesting facts about the experience of his organization with aluminum body tank trucks, pointing out that in the original design the chief effort was directed toward increasing the payload, and that only secondary consideration was given to reduction in body weight directly.

F. K. Glynn estimated that, during the life of a truck, $1.66 per lb. could be saved in operating costs through weight reduction. He pointed out, however, that the greater flexibility and the greater roadability to be obtained from lighter weight bodies was important as well as cost reduction.

The difficulty of arriving at a satisfactory uniform truck rating was emphasized by both M. C. Horine and B. B. Bachman, who have been on the S.A.E. Truck Rating Committee since it started its intensive work several years ago. Every type of rating devised, they indicated, tends to be unjust to one or another type of vehicle.

Reporting for a committee of the Transportation and Maintenance Activity which has been studying vehicle design from an operating and maintenance standpoint, F. L. Faulkner, Armour & Co., said, among other interesting things, "We do not advocate that all types of vehicles should be so closely standardized that they would lose their identity, but we do feel that if many items of design were standardized among the vehicle manufacturers, there would be a marked improvement in servicing costs." He said definitely that "there are entirely too many extreme cases of inaccessibility of parts for both minor and major servicing."

It is interesting to note that Mr. Faulkner did not lay all the blame for lack of uniformity of design on the truck manufacturer. He pointed clearly to the difference of opinion which exists among operators themselves as to what they want or need, adding that:

"We of the Committee recognize that our first big job among the operating personnel is to arrive at some agreement among ourselves as to what we need, and then only will we be able to influence the manufacturer to make the desired changes." He did mention 26 specific items, however, on which agreement had been reached by the important operators comprised in the committee.

The suggestions of the operators for possible changes in truck design were received with sincere appreciation by the many important motor truck engineers at the gatherings, and exchange of views regarding details went on for more than an hour.

From an economic standpoint, the factors affected by weight distribution of a motor vehicle, E. L. Tirrell pointed out, are the effect of tires, possibility of increased payload, steering, turning radius and reduced overall length. In the course of his thorough analysis of this whole subject, Mr. Tirrell brought out the fact that analysis of the increased payload possible to be obtained in 15 Eastern States reveals that in only 5 of these States is any advantage to be gained in redistributing weights from 25/75 to 33/67. In these 5 States the increased weights are: Delaware, 2000 lb.; New York, 3600 lb.; Pennsylvania, 2000 lb.; Virginia, 2700 lb.; West Virginia, 2700 lb.

He added, however, that this applies only to trucks which desire to carry the maximum loads permitted, and that such trucks are in a minority. "If a comparison between the ideal of the 33/67 were changed to 30/70," he said, "then the increased weights are considerably lower."

(Left to Right) T. P. Wright, speaker, Propeller Symposium; G. L. McCain, speaker, Streamlining Session; W. H. Graves, speaker, Passenger Car Fuels and Lubricants Session; E. P. Warner, chairman, Propeller Symposium

Every Day Was Ladies' Day at the Saranac Summer Meeting

1. Mrs. William H. Hubner
2. Miss Katherine Roos
3. Mrs. John C. Tuttle
4. Miss Betty Green
5. Miss Charlotte Anglada
6. Mrs. D. E. Gamble
7. Mrs. Harold Nutt
8. Mrs. Miriam Graham
9. Mrs. John R. MacGregor
10. Mrs. W. J. Cumming
11. Miss Peggy Marschner
12. Mrs. R. W. Davis
13. Mrs. Elbert E. Husted
14. Mrs. A. Ludlow Clayden
15. Mrs. R. E. Wilkin

Too Much Streamlining?

Most controversial of the conclusions arrived at by C. O. Guernsey, The J. G. Brill Co., in his discussion of design and application of motor trains, was that: "Streamlined motor trains have been carried to such a point in streamlining and structural materials that the cost is excessive. There would seem to be a need for compromise construction." In the light of current automotive and railroad interest in streamlined equipment, this particular statement of Mr. Guernsey's got particular attention from the 125 members who attended the session at which he talked.

W. B. Stout pointed out that the railroad business has been hampered in making desired changes in technical design and operating methods by political hindrances. They have to consult so many government agencies before making a change that by the time the politics have been worked out the chance to profit by the change often is passed.

A. K. Brumbaugh voiced the opinion that railroad executives very definitely are awakening to the possibility of change and that the railroads actually are making much progress despite the barriers which Mr. Stout had mentioned.

Walter Keys said that all the new streamlined trains start smoothly. It would be impossible, he said, to cut out a car from the articulated trains without tremendous difficulty. He voiced great admiration for the remarkable safety record which the railroads of America have made.

Mr. Guernsey said that his organization, The J. G. Brill Co., had built $18,000,000 worth of rail cars since first it entered that field of manufacture. Referring to Mr. Stout's comments, he said that he does not think that the engineering aspects of the railroad problem can be disassociated from the political and economic aspects.

In brief, the oil technologist's job is so to operate refining units as "to produce the most economical (Diesel) fuels at the lowest price and in the greatest quantity," Arch L. Foster said in his paper on the "Prospects for Future Diesel Fuels and Their Available Supply." He added that "The engine designer's problem is to design to utilize the most economical and most abundant fuel possible for the refiner to produce at the lowest cost; to adapt his products to take advantage of, rather than be handicapped by, the properties of the most economical fuels, to widen rather than restrict the choice of fuels available to the consumer." This was the thesis developed by the author and he indicated that it is the logical course to follow from an economic standpoint. Discussers found very little fault with these conclusions.

In their paper, "An Index of Diesel Fuel Performance," A. E. Becker and H. G. M. Fischer pointed out that they have found an excellent "correlation between the critical compression ratio and a function of analine point and gravity" of Diesel fuels. They say further: "Both of these physical constants are so readily determinable by simple and standardized laboratory methods that we are proposing this function be used as a 'Diesel Index' of ignition and combustion characteristics of fuel oils," the Index being equal to the analine point in deg. fahr. times the A.P.I. gravity divided by 100. The authors built a good case for this proposal, but some of the discussers expressed the view that further study of the proposal is necessary before it can be carried into practice or considered as a standard on which commercial specifications may be based.

George L. McCain, in presenting the first paper at the streamlining session said in part: "We . . . start with the rear passengers where they should be, build up a design around them and try at the same time to furnish comfort, stability and endurance." Wind tunnel tests showed that the factor K had been reduced from 0.00163 for the 1932 Chrysler sedan to 0.001375 for the streamlined Airflow sedan. By changes in weight distribution and in spring rate, "oscillation was brought down to about 90 cycles per min. without sacrificing the least bit of front-end stability." At the same time, riding properties were materially improved, as was illustrated by motion pictures of the old and new cars in operation over rough roads. Many other advantages were gained by the new type of construction developed.

Lowell H. Brown and Herbert Chase at the same session laid stress on the reduced wind resistance which the streamlined car has, but pointed out that several other important advantages, including better ventilation and cooling, can be gained. They showed the various models and cars developed by Jaray and concluded by saying, "If the information now available is put to proper use, all cars are likely to be well streamlined in the near future. Streamlining is amply justified on engineering grounds which cannot be ignored without impeding logical progress."

Several discussers, including W. B. Stout, Edward P. Warner and Louis Schwitzer took exception to some of the conclusions drawn and pointed out various difficulties likely to be encountered in streamlining, but the authors defended their position vigorously and showed in some instances that the discussers themselves were in error. Taken as a whole, the session was one of the best attended and most interesting of the meeting.

Diesel Design

F. M. Young presided at the Diesel Design Session on Wednesday morning which was well attended—despite fine weather after a day of rain. In the first paper, Torbjorn Dillstrom described the Hesselman spark-ignition fuel-injection oil engine and dealt with the reasons behind its design. He said that, although "this engine is a comparatively recent development, satisfactory results have been attained and it has been proved that the basic principles are sound. The engine has proved its practicability and reliability in several fields." It works on an efficient cycle and uses air efficiently. It can work on compression ratios of from 6 to 1 up to 10 to 1.

Two other papers presented at this session dealt with two quite similar methods of analysis of indicator diagrams in respect to combustion effectiveness. In the absence of J. C. Slonneger, author of one of these papers, both were presented by Hans Fischer, author of the second. Discussion indicated that the methods give promise of simplifying what has been a tedious and rather unsatisfactory procedure heretofore.

Those who took part in the discussion included F. C. Mock, Prof. L. C. Lichty, Harte Cooke and Arthur W. Pope, Jr. It was pointed out that absolute results are dependent upon the use of an indicator that gives accurate cards but that useful comparative results can often be obtained even when the indicator leaves something to be desired.

In closing the discussion, Mr. Fischer indicated that the method he developed has proved very useful. He also said that there is some tendency toward a return to air-injection types of Diesel engines in Europe, as they tend to reduce maximum pressures and thereby reduce bearing difficulties.

Stressing the point that the design of any object is inherent in the object itself, Walter Dorwin Teague said that the engineer produces beautiful results just in so far as he is successful in solving his engineering problem. He goes wrong when he tries to improve on the bare solution. That is why, in Mr.

(Continued on page 33)

The Summer Meeting in Pictures

1. Water sports got plenty of attention all week as well as on Monday afternoon when the Water Carnival was held.

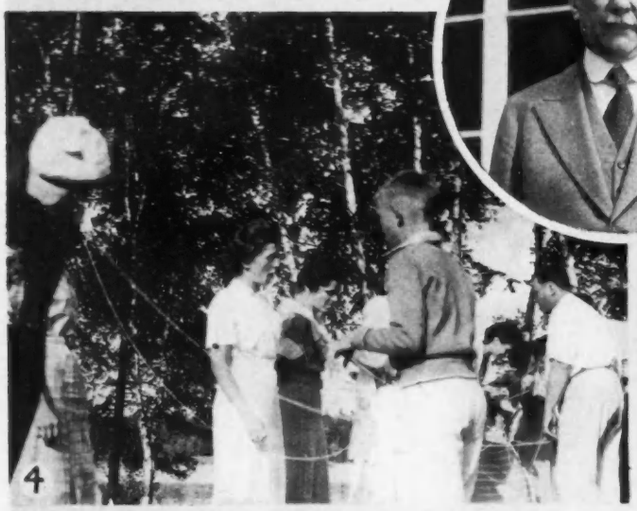
2. The Council stands "at rest" for a few minutes in the midst of a long meeting.

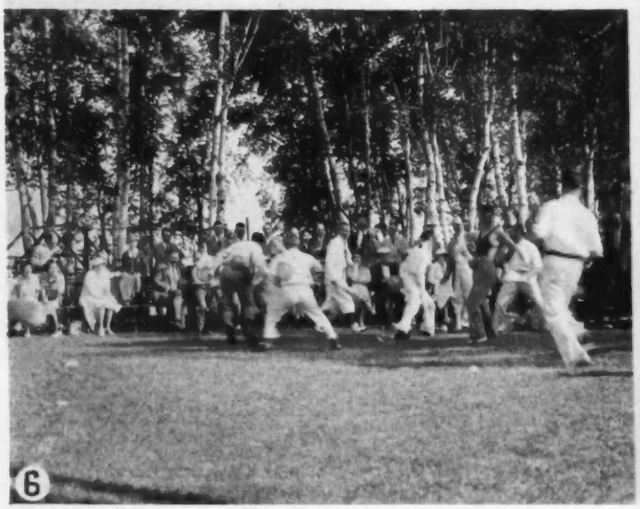
3. President Roos, Vice-President Robert Insley, General Manager John A. C. Warner, Councilor W. G. Wall and other Councilors await the beginning of the Sunday morning Council Meeting.

4. Some of the participants in the Water Carnival all ready to start.

5. Spectators crowded the dock for a close view of the Water Carnival events.

6. Trains arrived early at Saranac, but not earlier than the coffee which Saranac Inn Proprietor Harrington Mills had ready at the station for incoming guests.





1. A scene on the float during the Water Carnival.

2. The short 14th hole.

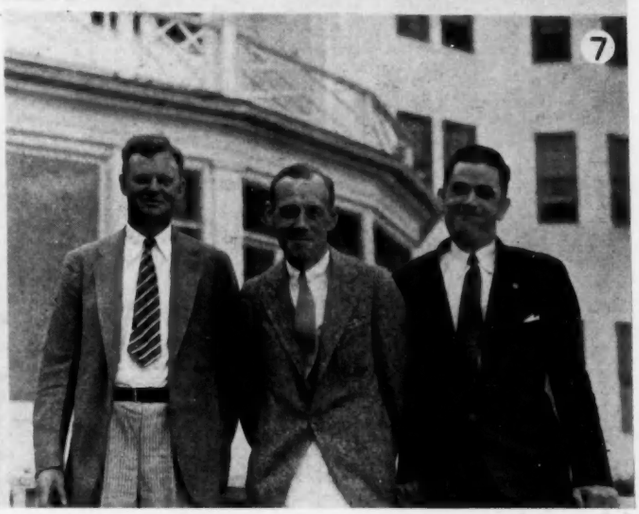
3. Walter C. Baker, one of the five life members of the S.A.E. and the first man in the United States to travel 60 m.p.h. in an automobile.

4. Another string-chewing Field Day event. If you look closely you will see why the finish was exciting.

5. Waiting to be "ringed".

6. A balloon on each back and a rolled-up newspaper in each pair of hands comprised the equipment for this most exciting of the Field Day events. The last man to preserve his balloon intact won.

7. A Field Day event for the gentler sex. The idea was *not* to have the bean bag in your hand when the whistle blew.



1. S. F. Dickey, generalissimo of the Field Day, calls for entrants to an event for ladies.

2. "I solemnly swear that I will finish this event . . ." chant the ladies before learning that they must lose their shoes and find them again.

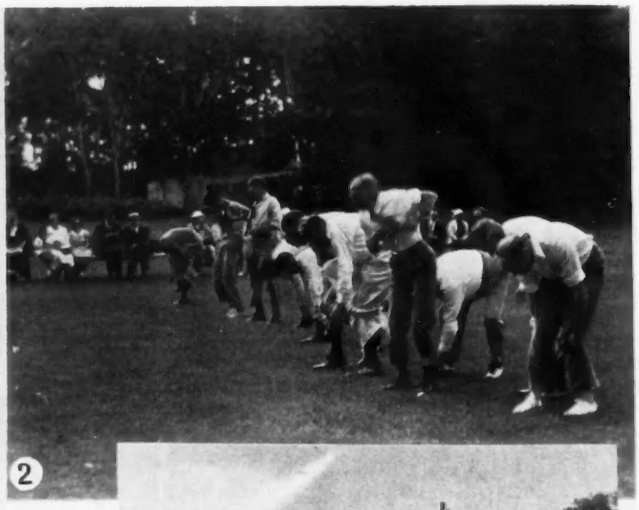
3. President D. G. Roos, once a newspaper photographer, shows that he has lost none of his cunning with the camera.

4. On the end of each string is a marshmallow. The idea is to see who can eat the string the fastest. Mrs. Tuttle, on the near end, did it.

5. A study in profiles—or is that the right word? Easily recognizable are A. K. Brumbaugh (left) and A. J. Scaife (right) with L. V. Newton just visible over the horizon.

6. The Metropolitan Section was well represented at Saranac. This picture includes some of its most prominent members who were in attendance.

7. C. Eustace Dwyer (left) was chairman of the golf committee. Here we see him with Past-President B. B. Bachman and L. Ray Buckendale.



1. The four-legged race was a Field Day feature.

2. Pull yourself through a rubber band; run a few steps, then pull yourself through another. Do this several times and you will know how it felt to be a participant in this particular Field Day event.

3. Past-President H. W. Alden and his daughter, Mrs. W. H. Hynard.

4. Vice-Presidents Kishline, McCoy and Insley.

5. Another view of the early morning coffee drinkers.

6. Saranac Inn from the lake.

7. Here's where the Water Carnival was held.

Summer Meeting Papers in Digest

HERE are digests of all the papers presented at the 1934 Summer Meeting. Next month will appear, in similar ready-reference form, digests of the discussion which took place following presentation of each of these papers.

* * * *

Some of these papers and discussions later will be printed in full in the S. A. E. JOURNAL.

Mimeographed copies of *all of them* will be available, until current supplies are exhausted, at a cost of 25c per copy to members; and at 50c per copy to non-members. Orders for mimeographed copies must be accompanied by remittance and should be addressed to Sessions Secretary, Society of Automotive Engineers, 29 West 39th St., New York.

Accident Prevention Session

Monday, June 18

Accident Control in Fleet Operation—*J. M. Orr, Equitable Auto Co.*

THIS paper is based upon the premise that the prevention and control of fleet vehicle accidents is an operating problem. Aside from the humanitarian aspects involved, the author says, fleet operators pay substantially for their accidents in settlement costs, interference with operations and generally lowered fleet efficiency.

Some of the questions he answers are how serious is this problem of accidents? How important is it to the average fleet operator? How much interest, time and effort is being taken and expended to improve present accident experience and arrive at an irreducible minimum?

Sound transportation and safety engineering are being successfully applied to accident control, the author states, which involves human engineering to a greater degree than in any other phase of fleet management and operation. We can definitely specify the types of vehicles used, and supervise their maintenance. We can select our operators very carefully, train them, restrain them, and what not, but the fact remains that drivers are self-supervised only while at the wheel.

Sound Session

Monday, June 18

The Place of Sound Measurements in Automotive Noise Reduction—*E. J. Abbott, University of Michigan.*

NOISE reduction must be judged in the final analysis by human ears, but ear observation is not an economical means of carrying on quieting work, this paper states. This is due primarily to the extremely wide range of intensity over which the ear must work, and the comparatively large change of intensity required to produce a perceptible change of loudness. This fact, coupled with the complex nature of most practical noises, usually makes the quieting obtained by a change in any part of a sound quite negligible. Hence ear observations do not furnish sufficient information on the relative importance of the various components of a sound, or the value of given quieting moves, which data are essential to logical and effective noise reduction.

The necessary data for carrying out effective noise reduction can be obtained by instrumental sound measurements, but the problem is not a simple one. The complications are not the fault of the meter, but arise directly from the complexities of practical noises, and the characteristics of the human sense of hearing. Successful noise reduction on the basis of measurements consequently involves both physical and psychological factors, and this paper outlines these two aspects of sound measurements as applied to automotive and machinery noise reduction, and indicates how these fundamentals are used in practical quieting.

Truck Design Session

Tuesday, June 19

Lightness in Truck Design—*Austin M. Wolf, Consulting Engineer.*

THE desire to save weight in a truck is similar to the endeavor in the airplane where every pound saved in the structure means that much more for payload. This paper is a survey of the field covering small, medium and large-sized trucks to indicate the general relationship between chassis and gross weights and the permissible weight for load.

The means of obtaining lightness in both chassis and body are (1)

the use of the proper alloy steels and materials in which the metallurgical aspect is as vital as the mechanical layout, (2) a close study of each individual part in the attempt to reduce its weight, and (3) the use of aluminum and duralumin.

The suggestion is made that in some instances the body and frame can be combined into a single structure. Standard structural members can be fabricated and stocked which will fit into varying conditions of length, width and height as well as body type.

The relationship of the weights of individual units to the total is considered and comments are made on the possible means of reducing the individual unit weights. A review is made of the late designs such as the camel-back and set-back front axle and their bearing on the subject of weight reduction.

Motor Vehicle Design Session

Tuesday, June 19

Vehicle Design from a Maintenance and Operating Standpoint—*F. L. Faulkner, Armour & Co.*

IT is the opinion among operators of larger fleets of motor vehicles that a material reduction in servicing costs could be made, provided vehicle manufacturers cooperated with them in effecting standards of such items as do not enter into their fundamental design, but which effect vitally the mechanical servicing of these units.

Sufficient data have been collected to indicate a wide variance of opinion among the fleet operators as to their actual needs. Likewise sufficient data have been accumulated from the manufacturers to indicate a lack of knowledge on their part of the average operator's requirements.

Numerous items requiring constant maintenance have been studied and recommendations outlined with a hope that it will ultimately lead to definite standardization.

The author anticipates this review of the situation will accomplish at least two things. First, a more unified opinion on the part of the operators as to their actual needs. Second, a better understanding and a more sympathetic attitude on the part of the manufacturer to the operator's requirements.

Weight Distribution of Motor Vehicles—*E. L. Tirrell, Utility Management Corp.*

THE purpose of the paper is to consider the advantages and disadvantages of various load distribution on tires, the possibility of increased payload, steering, turning radius and reduced overall length; for two-axle trucks requiring four tires, two-axle trucks requiring six tires, and three-axle trucks requiring ten tires.

The author also considers motorcoaches and cites the present practice of four different manufacturers in regard to weight distribution, and comments upon front-axle and engine location and steering ability.

Railcar Session

Tuesday, June 19

Trends in the Design and Application of Motor Trains —*Charles O. Guernsey, The J. G. Brill Co.*

THE history of the development of railcars or motor trains is reviewed and their economic application discussed. In the author's opinion, the eventual design for general application will be some reasonable compromise between the past conservative and the present extreme designs. His conclusions are:

(1) Rail motor-trains properly applied can be largely used to hold and regain traffic for the railroads.

(2) The principal competitor is the private automobile.

(3) Motor trains probably cannot compete with the private automobile or the common-carrier bus, in frequent-stop local-service.

(4) The cost of operating service per passenger seat should be much less than that of present equipment.

(5) Motor trains should be applied for operation over distances and at speeds where the overall time from point of origin to point of destination of the passenger will be less than by competing carriers, whether private or public.

(6) Frequency of service must be considered, as well as schedule speed.

(7) Because of speed, comfort and the like, such service should appeal to the public.

(8) Streamlined motor-trains have been carried to such a point in streamlining and structural materials that the cost is excessive. There would seem to be a need for compromise construction.

(9) The motor car more or less of past conventional types will continue to be used where the train consist must be varied.

(10) The power equipment will be either carburetor or Diesel type, depending upon the conditions of application.

Diesel Fuels Session

Tuesday, June 19

Prospects for Future Diesel Fuels and Their Available Supply—*Arch L. Foster, National Petroleum Publishing Co.*

IN view of the growing scarcity of Diesel fuels of low critical compression ratio, straight-run type, it appears, from a survey of the situation, that the Diesel of the future should be designed to utilize to the best advantage those fuels of the cracked and aromatic types which will undoubtedly be the cheapest and most-abundant fuels available.

A greater and greater proportion of the gasoline refined is obtained by cracking, especially by cracking the lighter distillate fuels. Curtailment of crude production, and therefore the amount of crude which may be processed, necessitates a greater amount of cracking to supply gasoline. Gas oils of the Diesel range are the premium charging stocks for cracking stills. Practically half the present gasoline supply is obtained by cracking.

Economic trends indicate that the most plentiful supply of fuels for both gasoline and compression-ignition engines will come via the cracking still, and/or the aromatic or naphthenic crude production. The two engines should not compete for the same source of supply. Straight-run gas-oil price will be determined largely by its value as a cracking-still charging-stock, i.e., its value as a gasoline source. To supply the present amount of straight-run gas oil from paraffinic sources, will necessitate the production of over half a billion barrels per year of straight-run heavy fuel oil, and the production of 2½ times the amount of crude now produced.

Economic trends and efficiency in engine development alike show that the compression-ignition engine should not become "temperamental about the type of fuel it needs."

An Index of Diesel-Fuel Performance—*A. E. Becker and H. G. M. Fischer, Standard Oil Development Co.*

ACCORDING to the authors, as the field for high-speed Diesel engines expands the question of suitable fuels and methods of rating them becomes of increased importance. That technical men are aware of this fact is clearly indicated by the increasing number of technical papers on the subject and by the recent organization of the Volunteer Group for C. I. Fuel Research, one immediate object being the development of a method for measuring the ignition quality of Diesel fuel with the expectation that it can later be recommended for universal adoption as a standard method.

In a previous paper referred to, Cooperative Fuel Research engine-data for 20 fuel oils were given, the results being expressed in terms of the critical compression ratio. Since that time Mr. Fischer has found that there is even better correlation between critical compression ratio and a function of aniline point and gravity. Both of these physical constants are so readily determinable by simple and standardized laboratory methods that the authors propose that this function be used as a "Diesel Index" of ignition and combustion characteristics of fuel oils, as defined by the equation

$$\text{Diesel Index} = \frac{\text{Aniline Point, Deg. Fahr.} \times \text{A.P.I. Gravity}}{100}$$

The authors then show the correlation of this "Index" with critical compression ratio, not only by use of the data in the previous paper mentioned but also for some 30 additional fuels.

Streamlining Session

Tuesday, June 19

Dynamics of the Modern Automobile—*G. L. McCain, Chrysler Corp.*

A NUMBER of conditions have contributed to airflow car development, this paper brings out. More rigid chassis construction, easy riding, and reclamation of formerly wasted space are of equal importance with streamlining and lessening of wind noise.

Limitation of car width being fixed by tread and tire size, the only way to gain passenger seat width was to move the seats to the wider section of the body and use the space in the rear for baggage. The result is checked by a simple apparatus showing the effect of the weight re-distribution. Moving pictures also show the action of the old and new suspensions.

High car speed which can be averaged with comfort has necessitated reduction of engine speed for passenger comfort, and fuel and oil economy. Any added speed which might result is secondary to the comfort involved.

The overdrive is an automatic high "high" and is controlled by a centrifugal clutch built into the transmission.

Streamlining—Up-to-date Facts and Developments—*L. H. Brown, Jaray Streamlining Corp., and Herbert Chase, Consulting Engineer*

STREAMLINING is not a mere style in flowing lines. It is, rather, a scientific development based on extensive tests which the optimum form for minimum air resistance in motor vehicles was determined with ground effect considered. Paul Jaray developed this form over twelve years ago and originated the term "streamlined car" to describe it. Though concessions to style and conventions are necessary, future designs should strive toward the ideal form for sound engineering reasons.

Results of the extensive Jaray tests on models and full-size cars are outlined and their economic and engineering significance are pointed out. There follows a list of advantages resulting from streamlining which, aside from reduced drag and power reduction, include ideal ventilation, superior engine cooling and good heating of body in winter.

Things to be done and to be avoided if air resistance is to be reduced to a minimum are pointed out. Wheels should be enclosed within the streamline fenders and running boards should be eliminated, boat-like shapes should be avoided and a longitudinal section approximating half of a true streamline form with flat bottom should be used. A rear engine mounting gives the best form but is not essential to good streamlining.

Historic as well as the latest streamline forms are illustrated and an extensive bibliography of articles on streamlining concludes the paper.

Diesel Design Session

Wednesday, June 20

Effective Combustion as Determined from the Indicator Diagram—*J. C. Slonneger, The Falk Corp.*

THIS paper sets forth a simple and rapid method of analyzing indicator diagrams with respect to effective combustion. Effective combustion is defined as combustion which actually increases the area of the indicator diagram over and above the combustion necessary to account for heat losses.

The method is based upon curves which are multiples of a re-expansion curve without ignition superimposed upon the diagram to be analyzed.

The author admits that this method is subject to criticism, but points out that the actual error is probably no greater than by other methods, and is perfectly satisfactory for comparing engines or checking laboratory results. It has the advantage of being rapid and simple in technique.

By this method the rate of effective combustion may be determined and thus assist the research engineer in checking the progress made in controlling combustion. This is especially applicable to Diesel type engines where the control of combustion is very essential.

Typical Indicator Diagram Analysis with Respect to Effective Combustion—*Hans Fischer, Lanova Corp.*

THIS paper points out that with the solid injection Diesel engine, the reliable smooth combustion performance of the old air-injection type has not been duplicated, especially so with the high-speed Diesel engine. To get the specific output as high as possible, and to obtain good fuel economy it is necessary to have the first part of the combustion approaching the constant volume-cycle, while the rest of the combustion is rather slow. In other words, the rate of burning is a maximum at the beginning and decreases toward the end of the combustion. The rate of burning in a gasoline engine is slow at the beginning and becomes a maximum at the end of the combustion, neglecting the slight after-burning.

Comparison is made with the rate of burning of different types of high-speed Diesel engines, with the one of the gasoline engine.

It is shown that, besides the hydraulic mixing of fuel and air an increased turbulence is required toward the end of the combustion, to obtain an efficient rate of burning.

A practical, desired indicator-card is shown, the increase of the rate of burning is comparatively small and the rate of pressure-rise is very moderate for the high specific output.

The Lanova engine is explained in which the burning takes place with increasing rate for 80 per cent of the fuel burned compared with the gasoline combustion in which 90 per cent of the fuel burns with increasing rate.

A High-Power Spark-Ignition Injection-Engine—*Torbjorn Dillstrom, Hesselman Motor Corp.*

THERE has, for many years, been a demand for an automotive engine more economical in operation than the gasoline engine. As the Diesel engine had early established a good reputation for economy, the development work on a more economical automotive engine quite naturally centered around this type of engine.

Additional advantages can, however, be gained by combining fuel injection with spark ignition, and an engine of this type has been developed by K. J. Hesselman of Stockholm, Sweden. This engine is generally called the Hesselman engine.

In the spark ignition fuel injection engine the charge is formed at a certain time before the spark occurs, and this makes it possible to mix fuel and air thoroughly whereby high output is secured. Engines operating on fuel oil with a compression ratio of 7.5 to 1 have given a brake mean effective pressure of over 125 lb. per sq. in.

The fuel is not injected until at the end of the compression stroke, and thus not unduly heated by the heat of compression. A higher compression ratio can therefore be employed in this engine than in the carburetor engine without pre-ignition or detonation. The engine thus works in an efficient cycle.

Oils of high thermal stability are difficult, and even impossible to burn in compression ignition engines. The local high temperature of the electric spark is, however, sufficient to start the ignition, even of a fuel with the highest thermal stability.

As these fuels are not liable to self-ignition, a very high compression ratio can be used if desired, and they therefore are to the Hesselman engine, what the premium gasolines are to the carburetor engine.

Body Design Session

Wednesday, June 20

The Basic Principles of Body Design—*Walter Dorwin Teague.*

THE design of an automobile body depends upon basic principles which are immutable, this paper states. These principles are applicable to all design and may be summarized as follows: Fitness to function, to materials and to construction. Unity, simplicity, proportion and line are also criteria which must be considered in judging the completed design.

Above these factors there is the innate ability of the designer which enables him to conceive fresh and unexpected forms and arrangements . . . gives his work that plus quality which can never be obtained by rule-of-thumb.

If the ablest designer departs from the principles, he errs, the paper states, and it can be demonstrated that every design which has achieved a striking commercial success has approached a realization of one or the other of the principles involved in the "perfect design."

All material objects may be seen in silhouette, the paper points out. Fender lines a few years ago were mostly bad, but have yielded to intelligent treatment. The rear lines of many of today's cars, as seen in silhouette, are bad also, Mr. Teague concludes.

Passenger Car Fuels and Lubricants Session

Wednesday, June 20

Winter Oils for Automobile Engines—*W. H. Graves, Packard Motor Co.; H. C. Mougey, General Motors Corp.; E. W. Upham, Chrysler Corp.*

IN this paper, it is brought out that the factors involved in cold starting of automobile engines, including the effects of temperature and oil viscosity on cranking speed and torque have been known for many years. Many papers have been presented before the various Sections of the S.A.E. on these subjects. The S.A.E. crankcase oil viscosity numbers, which were adopted in July, 1926, provided for the classification of the lower viscosity oils at 130 deg. fahr. and the higher viscosity oils at 210 deg. fahr. It was recognized by 1930 that a classification for winter oils must be based on the viscosity of the oil at the starting temperature and work was started on this problem. In June, 1933, the 10-W and 20-W oils, which are classified in accord with their viscosity at 0 deg. fahr. were adopted for publication and trial.

The results of the use of these oils during the winter of 1933-34, together with their advantages, are discussed. The data show that these oils are necessary for cold starting, provide adequate lubrication and improved performance and give reasonable oil mileage. Although higher viscosity oils give more miles per gallon of oil, by far the most important factor in oil consumption is engine speed. The lower viscosity oils, that are required for winter operation, reduce engine friction and the total cost of oil plus gasoline using oils of lower viscosity may be less than when using oils of higher viscosity.

Fuel Characteristics and Vapor Lock—*O. C. Bridgeman, Bureau of Standards.*

IN the last previous report to the Society on the investigation of vapor lock, a number of predictions were made regarding the effect of fuel characteristics on vapor lock. In order to test these predictions, vapor lock road tests were made with 18 different cars using fuels of widely differing characteristics. The results of these tests, discussed in this paper, verified the predictions, namely, that from road tests with one type of fuel it is possible to evaluate with sufficient accuracy the conditions under which other types of fuel will vapor lock in the same car.

For any given car, it was found that, in general, a fuel with a flat distillation curve caused vapor lock more readily than a fuel with a steep distillation curve, the difference being more marked if the car has a high vapor-handling capacity. However, when considering all of the cars on the road at present, the permissible vapor pressures for reasonable freedom from vapor lock are independent of the fuel characteristics. The reasons for this are two-fold. In the first place, most of the cars having high vapor-handling capacity will not vapor lock on any of the commercial fuels, and even a marked change in fuel characteristics will not cause vapor lock if the vapor pressure is within the present commercial range. In the second place, those cars which give most trouble from vapor lock have low vapor-handling capacities and under these conditions, differences in fuel characteristics have little effect. Accordingly the vapor pressure limits previously suggested as giving reasonable freedom from vapor lock, are still considered to be satisfactory regardless of the distillation characteristics of the fuel used.

Bearings and Lubrication Session

Wednesday, June 20

Recent Developments in Main and Connecting Rod Bearings—*Stanwood W. Sparrow, The Studebaker Corp.*

UNTIL recently babbitt has been used almost exclusively in main and connecting rod bearings. Cracking of the babbitt usually is the initial cause of failure. Such failures may be due to flexing or faulty bonding. In this paper, experiments are described showing that these failures may be due to the breaking of the babbitt at the surface of the bearing. Evidence is presented which shows that cracks usually start where the journal breaks through the oil film and the tangential force exerted by the shaft is suggested as the cause of failure. Failures which result from high temperatures are attributed to inadequate lubrication rather than to changes in the properties of the bearing material. Where it is not possible to prevent babbitt from cracking it is pointed out that satisfactory life can be obtained by expedients which prevent the broken babbitt from escaping.

Copper-lead mixtures have been found to give long life under severe service. Troubles which have been experienced with this type of bearing are described and considerable attention is devoted to failures resulting from the use of unsuitable oils.

Problems in lubrication which have arisen as the result of high engine speeds are discussed and emphasis is placed upon the importance of getting an adequate amount of oil to the connecting rod bearing and of preventing abnormally high operating temperatures.

Lubrication of Engines with Different Bearing Metals, with Special Reference to Copper Lead Alloys—*C. M. Larson, Sinclair Refining Co.*

WITH the advent of light motor oils, S.A.E. 10-W and S.A.E. 20-W and the introduction of copper lead bearings in passenger car and motor truck engines, new lubrication factors have arisen with respect to motor oil stability as well as with respect to the nature of compounds used for the breaking-in period.

This paper describes a series of tests which were run using S.A.E. No. 11 babbitt bearings and lead copper and leaded bronze bearings. The relative extreme-pressure characteristics of the various motor oils,

both mineral and compounded were recorded. In addition, the coefficients of friction were plotted as ordinates (f) and the viscosity, angular speed and pressure ($Z\ N/P$) as abscissae were correlated. For this phase of the work, test apparatus had to be developed.

The effect of the mineral and compounded motor oils on the bearing metals and the causes of deterioration, and vice versa, the effect of the bearing metals on the stability of the motor oils were checked.

The summation of the data shows that certain metals cause some mineral oils to deteriorate more readily than others and that high acidity in motor oils causes not only more rapid decomposition of the lubricants themselves but of the bearing metals as well. The coefficient of friction varies with the different metals and with the different combinations of compounds.

Engine Session

Thursday, June 21

Engine Types Adapted to Automobile Design Trends —*Lewis P. Kalb, Continental Motors Corp.*

THE American automobile industry has adhered to the in-line and V-type engines exclusively for practically all of its mature years. The suitability of these conventional engines to the structure and arrangement of the automobile is due primarily to the fact that engines and vehicles have grown up together. At the present time radical changes in automobile structure seem to be in the offing. In view of this, it is probable that changes in engine arrangement are also imminent.

This paper discusses unusual forms of engines and compares their adaptability to the newer types of automobile with the conventional engines now in general use. Discussion has been confined to those engines for which there is some background of successful application, either in the automobile or in some kindred automotive field. The types discussed include: The in-line engine, the V-type engine, the flat engine, the barrel engine and the radial engine.

The advantages and disadvantages of these various engines are discussed with respect to ease of adaptation to various types of cars, balance, even firing, torsional conditions, weight, compactness, ease of manifolding, accessibility and ease of manufacture. It is brought out that the 4-cylinder opposed engine has many attractive features which would lend itself to a low-cost automobile. The attractive possibilities of adapting the radial engine to the rear-engine type of car also are pointed out.

Possibilities of Forced Induction in Automotive Vehicles—*Louis Schwitzer, Schwitzer-Cummins Co.*

IT is always the automotive engineer's endeavor to improve automotive vehicular performance. Greater acceleration and higher speeds have been accomplished by a continuous increase of engine sizes, until today further increase by this method is not logical.

We are today turning to dual axle ratios and streamlining to accomplish further increases; the former to allay the disastrous effects of the drooping torque curve at high engine speeds, and the latter to reduce the power to propel the vehicle. Reduction of engine speed-wheel speed ratios is ideal for high speed but represents a loss in acceleration at intermediate speeds. Reduction of wind resistance of the vehicle, only valuable at high speeds, permits only a slight increase in top speed because of the rapid reduction in power at high motor speeds. Thus a dilemma exists, which can be avoided by forced induction, or supercharging. Forced induction with dual axle ratios and streamlining produces the only complete picture.

There are three general types of compressors adaptable to automotive engines. They are the vane, Roots, and centrifugal types. The latter is best adapted for present-day engines. Actual tests show, in addition to a remarkable improvement in performance, a sizable reduction in specific fuel consumption.

Centrifugal compressors are least expensive, most silent, involve the fewest problems, have the greatest production possibilities, and can be adapted to present engines in a number of satisfactory ways.

Propeller Symposium Session

Thursday, June 21

Recent Developments in Aircraft Propellers—*F. W. Caldwell, Hamilton Standard Propeller Co.*

THIS paper treats in a very brief way with the aerodynamics of the propeller including the effect of selection of the airfoil, section, the planform and the aspect ratio. The aerodynamic characteristics of the propeller as a whole are discussed very briefly with some reference to the effect of tip speed and body interference.

The stress analysis is discussed briefly and methods of measuring the vibration stresses are described in some detail.

There is a short analysis of the properties and advantages of some of the materials used for propeller blades.

A short outline of the development of the controllable pitch propeller follows with a description of the present type of hydro-controllable propeller now in use all over the world.

The last part of the paper treats methods of testing propellers to insure their safety in service.

Design Requirements of Mechanical Controllable Propellers—*G. T. Lampton, Lycoming Mfg. Co.*

RAPID expansion of the controllable propeller market makes desirable a general review of the design conditions to insure the strength of the production articles in all flight regimes. Fortunately the external loads on a family of propellers can readily be calculated from the tip speed limitation. This determines the r.p.m. of extreme diameters of all basic blade designs, and, with assumption of engine and airplane performance guided by actual data on modern combinations, the worst probable external propeller loads can be evaluated. The dive is the general design condition, as both the loads and moments are maximum in this flight regime.

Due to the severe stresses to which propellers are subjected and the requirement that all parts be interchangeable, close tolerances are necessary. The comprehensive tooling to achieve this requires close coordination between production and engineering departments.

Controllable Pitch Propellers—Design Considerations —*T. P. Wright, Curtiss Aeroplane & Motor Co., Inc.*

IN this paper the author has first shown the increasing need, amounting in many instances to a necessity, for the adoption of the controllable pitch propeller on modern airplanes. Improvements in airplane design which bring about increased speed range have brought about a decrease in other performance items which can only be rectified by the use of the controllable pitch propeller. The improvement has been shown to be so great as to pay well for the added complication and expense involved.

The several types of controllable pitch propellers developed in this country are described, the author going into considerable detail on the electric-driven type.

The numerous test requirements are discussed in detail. In conclusion, the author notes greater progress in the development of the controllable pitch propeller in this country than elsewhere perhaps necessitated by the accompanying advance in aerodynamic refinements here developed.

Bearings for Controllable Propellers—*Thomas Barish, Marlin-Rockwell Corp.*

THE controllable propeller involves a difficult and unusual bearing problem.

There are enormous centrifugal forces due to the tendency of the blade to fly outward and large sideways forces because the bearings are so close together at the hub. These must be carried with a minimum of friction to permit easier movement of the blade when changing the angle.

In addition, the weight and outside diameter of the bearing must be kept as small as possible. Hence the new concept of "Brinelling capacity" representing the maximum static load that can be sustained before any permanent deformation of the bearing surfaces occurs.

This paper outlines the various methods used to solve this bearing problem and some of the tests and comparisons made in arriving at these designs.

Automatic Variable-Pitch Propellers — *Clinton H. Havill, Eclipse Aviation Corp.*

VARIABLE thrust or drawbar pull is really the fundamental factor upon which propeller performance is based, according to the author; therefore, any propeller has only one duty, which is to convert power into thrust at various airspeeds. Since thrust combines power, airspeed and altitude, as well as propeller speed, diameter and pitch, it seemed logical to design a propeller structure in which the pitch is varied according to the amount of thrust, but this constituted a difficult problem because the variables that enter into thrust are each, in turn, functions of the others. Hence, a knowledge of the rate of change of the variables became necessary.

An estimate of the five simultaneous differential equations involved indicated that a structure built to link thrust and pitch was feasible, and such a propeller was built. It was used to obtain the necessary quantitative data. Later redesign and further experimentation resulted in a satisfactory propeller, its operation being based on the premises that (*a*) an increase of thrust decreases pitch and that (*b*) a decrease of thrust increases pitch.

The propeller structure is illustrated diagrammatically and explained. The propeller is mounted in the same way as for any conventional American propeller, and can be installed or removed as a unit. Its three adjustments can be made without removing the propeller, and there are no controls from the cockpit.

Design and Combustion Session

Thursday, June 21

Observations of Flame in an Engine—*Charles F. Marvin, Jr., Bureau of Standards*

DURING an investigation conducted at the Bureau of Standards under the sponsorship of the National Advisory Committee for Aeronautics, visual and photographic observations have been made of the spread of flame to all parts of the combustion chamber of a single-cylinder L-head engine. Heads equipped with a large number of small windows symmetrically distributed over the combustion chamber were observed through a stroboscope, flame diagrams being obtained for a wide range of engine operating conditions and for a variety of fuels, combustion chamber shapes, and arrangements of single and twin ignition. In this paper, the major factors influencing flame movement in the engine are discussed and their effects upon the diagrams are indicated.

As a means of studying more intensively conditions in and behind the flame front, measurements were made of the variations in intensity and spectral distribution of the infra-red radiation emitted through selected windows during normal and knocking explosions. The character of this radiation indicates that it is emitted almost exclusively by the final reaction products, water vapor and carbon dioxide and the variations in its intensity show that reactions producing these substances continue in a normal explosion of 20 deg. or more of crank angle after inflammation of an element of charge. In those portions of the charge where fuel knock occurs the reactions appear to be much more rapid.

Design Limitations of Aircraft Engines—*E. S. Taylor, Massachusetts Institute of Technology*

IN this paper, the general design problem is considered by reference to the principle of similarity.

The principal is briefly reviewed and applied to lubrication, propeller drives, cooling and detonation.

It is shown that while small engines should be capable of a larger output per square inch of piston area, in practice the reverse is the case. It is also shown that small engines should, in general, operate at a higher efficiency than large ones. A promising line of attack for improving the lubrication of small engines is indicated.

Cooling and Detonation Research Session

Friday, June 22

The Design of Fins and Cylinder Baffles for Air-Cooled Engines—*Carlton Kemper, National Advisory Committee for Aeronautics.*

THE National Advisory Committee for Aeronautics has undertaken a research to determine the effect of the fundamental factors influencing the cooling of air-cooled radial engines. This paper presents some of the results of an investigation made to obtain information that can be used in the design of fins for the cooling of heated cylindrical surfaces in an air stream. An equation has been developed for calculating the quantity of heat dissipated by a given fin design using experimentally determined heat-transfer coefficients. The agreement between the quantity of heat dissipated as determined by experiment and calculated by the equation was found to be sufficiently close to justify its use.

A method has been developed for determining the fin dimensions utilizing a minimum of material for a variety of conditions of heat transfer, air flow, and metals. The design of cylinder baffles has also been investigated and results are presented that show the improvement in cylinder temperature distribution and the increase in the average heat-transfer coefficient to be obtained by the use of shell and integral cylinder baffles.

Correlation of Knock Rating of Aviation Gasolines—*Arthur Nutt, Wright Aeronautical Corp.*

THIS paper reviews briefly the history of the C.F.R. Aviation Gasoline Detonation Subcommittee. It outlines the problem, the program of testing three types of fuels, namely, (1) fuels giving constant values under a variety of test conditions; (2) fuels showing depreciating values as test conditions became more severe; (3) fuels showing appreciating values as test conditions became more severe. The test procedure is outlined with a brief statement on the actual test work. A discussion of the results follows stating that while no definite conclusions can be drawn when the work is only 42 per cent completed yet trends are evident.

The straight run gasolines show no deviation from C.F.R. Motor Methods. The benzol blends show a depreciation particularly in the higher octane numbers on the larger engines. The cracked gasolines show a limited depreciation on the smaller engines in the lower octane numbers. In the larger engines some depreciation is noted with the Hornet and Wasp engines, but none in the large liquid cooled engine (Conqueror). No data are available on the action of the appreciating types of fuels.

A method of determining a match number is suggested which may be useful as a quantitative measurement.

The investigation so far appears to be leading to results which will make possible recommendations for changes to the C.F.R. Motor Method of checking fuels so that a more suitable method of rating fuels for aircraft spark-ignition engines will be available.

Shock Absorber Session

Friday, June 22

The Airplane Landing-Gear Shock-Absorbing System—*C. V. Johnson, Bendix Products Corp.*

THE shock-absorbing system of the main landing gear of an airplane must function under the impact of landing and while taxiing on landing fields, this paper points out. Present requirements for impact landing are outlined, and a typical analysis made to check up a proposed system for a given airplane. Effects of geometrical arrangement of gear, of tire size and of tire rebound are considered. Laboratory methods of testing systems to determine if requirements are met are discussed.

The characteristics desired of the shock-absorber system for good taxiing are enumerated, and effects of various types and arrangements of absorber units on performances investigated. The paper is concluded by a discussion of the tail-wheel shock-absorber system in which it is brought out that the same criteria that are applied to the main gear apply here, but that the relative importance of various factors is not the same as in the main gear.

What Members Are Doing

B. B. Bachman, David Beecroft, H. C. Dickinson and *A. J. Scaife,* all past-presidents of the Society, and *A. K. Brumbaugh,* vice-president representing Truck, Bus and Railcar Engineering, were among the official representatives of the Society at the National Conference on Street and Highway Safety held in Washington at the end of May. Several of the representatives participated in committee meetings prior to the general conference.

John H. Geisse is chief of the manufacturing section of the Aeronautics Branch, Department of Commerce, Washington.

J. H. Geisse

Louis A. Gilmer has joined the Oliver Farm Equipment Co., Charles City, Iowa, as designing engineer. He was connected with the John Deere Tractor Co., Waterloo, Iowa.

Raymond G. Perry has been named passenger traffic manager of the Provincial Transport Co. in addition to his duties as general manager of the Colonial Coach Lines, Ltd. His address is 1227 Phillips Sq., Montreal.

Thomas L. Hibbard has formed the Thomas L. Hibbard Associates, an industrial designing organization, in New York City. Mr. Hibbard was formerly designer for the Fisher Body Corp., Detroit.

Robert S. Merithew is a salesman with the M & M Motor Sales, Inc., Norwich, N. Y.

Elwell R. Jackson is manager, automotive maintenance division, Robert S. Breyer Co., Ltd., Los Angeles.

E. R. Birchard has been named assistant general sales manager of General Motors Products of Canada, Ltd., after service as zone manager in the Toronto area for the same organization.

W. G. Nagel is president of the Air Conditioning Corp., Detroit, recently formed to sell General Electric air-conditioning products.

Adolph A. Blattner has formed an organization in Pittsburgh to sell Ford cars and trucks and Linn Tractors. The company will be called Adolph A. Blattner, Inc.

Mason D. Hanes has joined the General Motors Truck Co., Pontiac, Mich., as design engineer. Since the first of the year, Mr. Hanes had been design engineer with the Lycoming Mfg. Co., Williamsport, Pa.

W. Fairhurst whose election as vice-president in charge of sales, Spicer Mfg. Co., was announced last month

Albert Vere Oliver has joined the Daimler Co. at Coventry, England, and is doing experimental design engineering.

Chester R. Wells, formerly design engineer in the brake division, Stewart-Warner Corp., Chicago, has joined the Pierce-Arrow office of the Patent Engineering Corp., Buffalo, N. Y.

C. O. Larson has left the employ of the Eddy Match Co., Pembroke, Ont., to return to Sweden, where his address will be Bangatan 8, Gothenburg. He was mechanical engineer and superintendent of the Eddy organization.

William H. Bassett, metallurgical manager, American Brass Co., has been nominated for president of the American Society for Testing Materials. He will serve for one year.

J. B. Rather, who is in charge of the general laboratories, Socony-Vacuum Corp., has been nominated as a director of the A.S.T.M. for two years.

William Ernest England, who was for 9 years chief engineer of the F. B. Stearns Co., producers of Stearns-Knight automobiles in Cleveland, has been named chief engineer of the Ohio Rubber Co., Willoughby, Ohio. Recently Mr. England has been associated with the Auburn Automobile Co., as engineer in charge of the experimental department.

W. E. England

Wide World Photo

Frank W. Caldwell, research engineer, Hamilton Standard Propeller Co., received the Collier Trophy for 1934, at a presentation made in Washington by President Roosevelt on May 31. The Trophy is awarded annually for "the greatest achievement in aviation in America" during the year previous to that in which the award is made. Mr. Caldwell, who has been a member of the Society since 1917, received the Trophy for his invention of a controllable-pitch aircraft propeller. He presented a paper on the subject at the Summer Meeting of the Society at Saranac Inn, N. Y.

Wide World Photo

Isabel C. Ebel

Miss Isabel C. Ebel, who joined the Society as a junior member in 1933, has been awarded the degree of Aeronautical Engineer by the Daniel Guggenheim School of Aeronautics at New York University. Miss Ebel is believed to be the first woman in the United States to be awarded the degree. Her thesis was a design for a "flying-wing" type of airplane.

In a newspaper interview regarding her choice of vocation, Miss Ebel said: "Up to now, woman's place in aeronautics has been limited to positions in servile capacities, such as hostesses and secretaries. Whether women can succeed in the aerodynamic phase of the industry is, of course, questionable, but there must certainly be room for them in another important side of the business.

"About 50 per cent of the passengers on transport planes are women and since that is the case, the commercial airlines must create a feminine appeal. It is here, I believe, that the woman designer fits in, not only in the interior designing of the plane itself, but in the design of airports and other related services."

Arthur M. Marshall has joined the M & R Transportation Co., Inc., Springfield. Mass. He was formerly assistant manager of the Malkin Motor Freight Co., Cambridge.

B. J. Lemon's new address is c/o U. S. Rubber Products, Inc., 440 W. Washington St., Chicago.

Frank B. Averill is production manager for the Studebaker Corp. of Canada, Ltd., Walkerville, Ont. He was factory manager for Dominion Motors, Ltd.

James F. Knudson, Jr., is design draftsman with the Brewster Aeronautical Corp., Long Island City, New York. He was formerly with the Curtiss Aeroplane & Motor Co., Kenmore, N. Y.

William F. Barge, formerly assistant master mechanic of the Lycoming Mfg. Co., Williamsport, Pa., has been made chief tool engineer.

Steven F. Evelyn is consulting engineer to the air-equipment division of the Worthington Pump and Machinery Corp., Harrison, N. J. Mr. Evelyn was formerly chief engineer on industrial engines for the Continental Motors Corp., Detroit.

Robert W. Wilson, who was vice-president of the Perfex Corp., Milwaukee, continues as vice-president and director of the Perfex Radiator Co., which has been formed to take over the business and assets of the Perfex Corp.

Ferdinand A. Bower, chief engineer, Buick Motor Co., received the degree of Doctor of Science from Villanova College, Villanova, Pa., during the commencement ceremonies

Ferdinand A. Bower

at the College on June 7. The honorary degree was awarded in recognition of Mr. Bower's achievements as an automobile designer during a career in the industry which began in 1908. He was named chief engineer of Buick in 1928, after serving 10 years as assistant chief engineer. He has been a member of the Society since 1912.

Stuart G. Baits, chief engineer of the Hudson Motor Car Co., has been appointed assistant general manager. He will continue to function as chief engineer.

Walter O. Will has been appointed toolroom foreman of the Union Special Machine Co., Chicago.

Keith S. Duncan, formerly assistant rolling-stock superintendent, Municipal Tramways Trust, Adelaide, So. Australia, has joined the Industrial Service Engineers Pty., Ltd., West Melbourne, Victoria.

Arthur L. McCullough has been on assignment in Panama for Pan American Airways. He was manager of the Compania Nacional Cubana de Aviacion before joining Pan American.

Louis H. Palmer

Louis Hooker Palmer, vice-president and general manager, Fifth Avenue Coach Co., New York City, died May 31 after an illness of several weeks. He had been identified with transportation interests throughout his business career and joined the Society in 1924, when he was vice-president and general manager of the United Railway & Electric Co., Baltimore.

From 1926 to 1932, Mr. Palmer was a member of the Motorcoach Division of the Standards Committee. Other S. A. E. committees on which he served included the Operation and Maintenance Committee and the Transportation and Maintenance Activity Committee. He had been the S. A. E. representative on the Committee on Motorcoaches of the American Electric Railway Association and on an American Transit Association Subcommittee on Motorcoaches. In 1922 he served as president of the American Electric Railway Association.

Mr. Palmer is credited with being a pioneer in the coordination and development of motorcoach operation in urban transportation. He was one of the presidential appointees to the National Conference on Street and Highway Safety and helped to organize the Baltimore Safety Council.

He was born in Chicago in 1880 and attended the Oak Park Illinois High School and Williams College. He began his business career as assistant chief clerk in the Jersey City offices of the

Buchrach

Louis H. Palmer

Central Railroad of New Jersey. In 1906 he became assistant to the general manager of the Metropolitan Street Railways and the New York Railways Co. and his subsequent career was entirely in the field of urban passenger transportation.

Reginald G. Burr

Reginald G. Burr, representative in St. Louis of *Bus Transportation*, died June 2 of acute appendicitis. He had taken an active interest in the formation of the probationary Section in St. Louis. Mr. Burr, who was 35 at the time of his death, had joined the Society as a member in 1931.

Obituary

Deaths of the following members of the Society have been reported to headquarters during the month preceding this issue of the JOURNAL: John R. Rogers, consulting engineer, Mergenthaler Linotype Co.; Charles D. Shain, Layton W. Averill, and Claude A. Wales.

Recent Developments in Main and Connecting-Rod Bearings

By Stanwood W. Sparrow

Studebaker Corporation of America

MR. SPARROW states that the widespread use of babbitt as a bearing material may be due to the fact that metal-to-metal contact over a small portion of the surface causes the material to soften and crush without damaging the journal or communicating sufficient heat to the remainder of the bearing to destroy the oil film.

To some extent copper-lead mixtures are superseding babbitt for main and connecting-rod bearings, and Mr. Sparrow discusses these primarily from the standpoint of their performance in comparison with that of babbitt.

Regarding both bearing materials comments are made upon such subjects as cracking, bonding, thickness, life, overheating, disintegration, lubrication and the use of suitable oils, and the results of numerous tests are presented.

THIS paper deals only with main and connecting-rod bearings, and its discussion of bearing materials is confined to babbitt and copper-lead mixtures.

Babbitt

Experience has made the automotive engineer a pessimist. He designs his lubricating system to maintain a film of oil between the journal and bearing but realizes that he will not succeed and takes care to select a bearing material which will minimize the harmful effects of his failure. Babbitt is the material which for many years has seemed best suited to this purpose. Babbitts differ widely as to analysis and physical properties. In one engine the best results were obtained with a babbitt consisting of 2½ per cent of copper, 8 per cent of antimony and 89½ per cent of tin. In another engine satisfactory results were obtained with 3.7 per cent of copper, 7.3 per cent of antimony and 89 per cent of tin. Results very nearly the same were obtained with 4½ per cent of copper, 4½ per cent of antimony and 91 per cent of tin. Despite a considerable amount of test data, it would require some courage to state positively that any one of these three analyses is definitely superior to the others. The widespread use of babbitt may be due to the fact that metal-to-metal contact over a small portion of the surface causes the bearing material to soften and crush without damaging the journal or communicating sufficient heat to the remainder of the bearing to destroy the oil film.

Cracking.—The foregoing statements apply particularly to failures in which sufficient heat in generated to melt the bearing. At present, the initial cause of failure is more often the cracking of the babbitt as illustrated in Fig. 1.

Bond.—It goes without saying that there should be a good

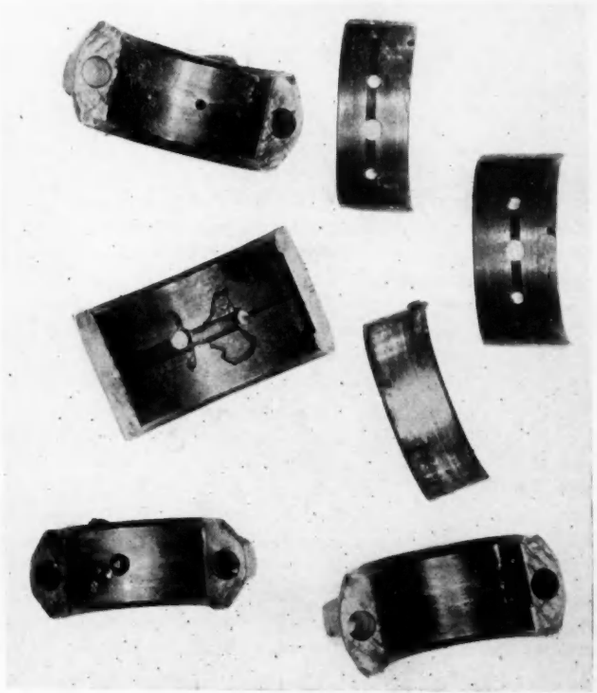


Fig. 1—Typical Examples of Cracked Babbitt

[This paper was presented at the Semi-Annual Meeting of the Society, Saranac Inn, New York, June, 1934.]

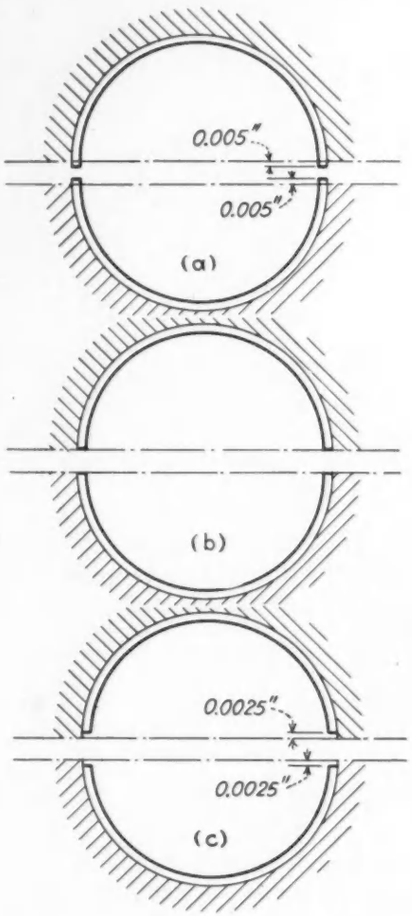


Fig. 2—Diagram Showing That Changing the Parting-Line Height of Thin-Back Bearings Did Not Affect the Tendency of the Babbitt to Crack

bond between the babbitt and the connecting-rod or bearing shell. Fortunately, the importance of clean metal surfaces and of the temperatures of babbitt and shell are well understood and defects in bonding are more often due to faulty control of production than to lack of knowledge. In this connection mention might be made of one instance where a presumably satisfactory cleaning solution was found to contain copper due to the presence of a copper washer in the dipping apparatus. The amount of copper was small, but the result was disastrous.

Deflection of the Bearing Shell.—There have been numerous instances where babbitt has cracked as the result of repeated deflections of a weak section of a connecting rod and where this trouble has disappeared when the rigidity was increased. In the initial tests with thin-back bearings (bearings having a thickness of shell not exceeding 1/16 in.) it was thought that an incorrect parting-line height might permit this breathing action and thus contribute to bearing failure. Anxiety on this point was dispelled by a series of tests under three conditions of "crush"; namely, (*a*) with total circumference of bearing shells greater than the circumference of the crankcase hole by 0.020; (*b*) with total circumference of bearing shells equal to the circumference of the crankcase hole; and (*c*) with the circumference of bearing shells less than the circumference of the bearing hole by 0.010 in. These three conditions are indicated in Fig. 2. The tests showed no difference between the three sets of bearings as regards cracking. This does not mean that the matter of "crush" is unimportant, but merely that it does not affect the flexing of the liners sufficiently to produce any significant difference in bearing life.

Temperature.—There are many bearing failures which can be traced directly to operation at abnormally high temperatures. From time to time reports have been published which show changes in the physical properties of babbitts with change in temperature and these seem of sufficient magnitude to explain the bearing failures. Nevertheless, in the cases with which I am familiar, the primary cause of failure has been the reduction in the amount and viscosity of the oil supplied to the bearings as a consequence of the high temperatures rather than to any change in the characteristics of the bearing material. This will be discussed later in connection with the subject of lubrication. Where this is true it is evident that developments which minimize the effect of temperatures upon the properties of bearing materials will be of minor importance in preventing failure.

Some evidence along this line is furnished by Fig. 3. This shows connecting-rod bearings after 50 hr. of full-load operation with crankcase-oil temperatures of 140, 180 and 300 deg. fahr. The bearings show a negligible difference in appearance, although the factor of safety against failure is not great as evidenced by the appearance of a similar set of bearings after a test with the higher inertia loads resulting from the employment of pistons of cast iron instead of aluminum.

Another example, less definite but nevertheless of interest, is furnished by Fig. 4. In this case both sets of bearings were

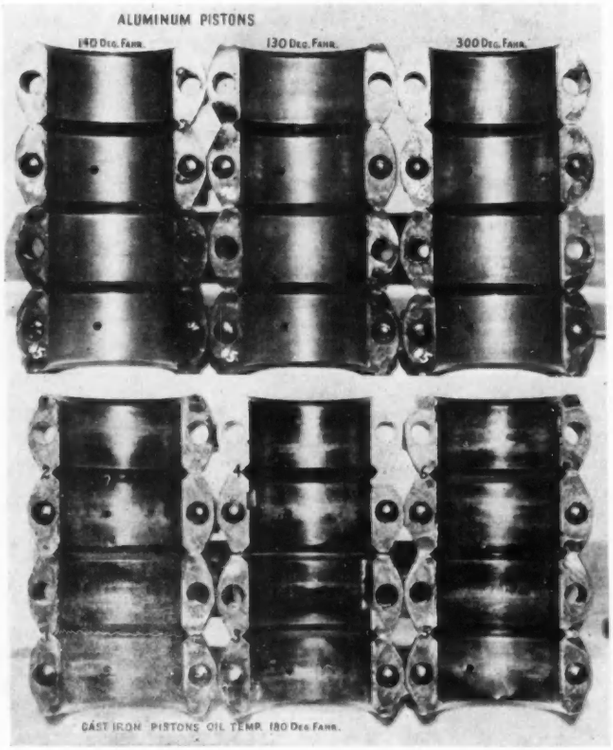


Fig. 3—Illustrating That, in The Tests of 50 Hr. at 4000 R.P.M., an Increase in Oil Temperature Was Much Less Harmful Than an Increase in Inertia Loads

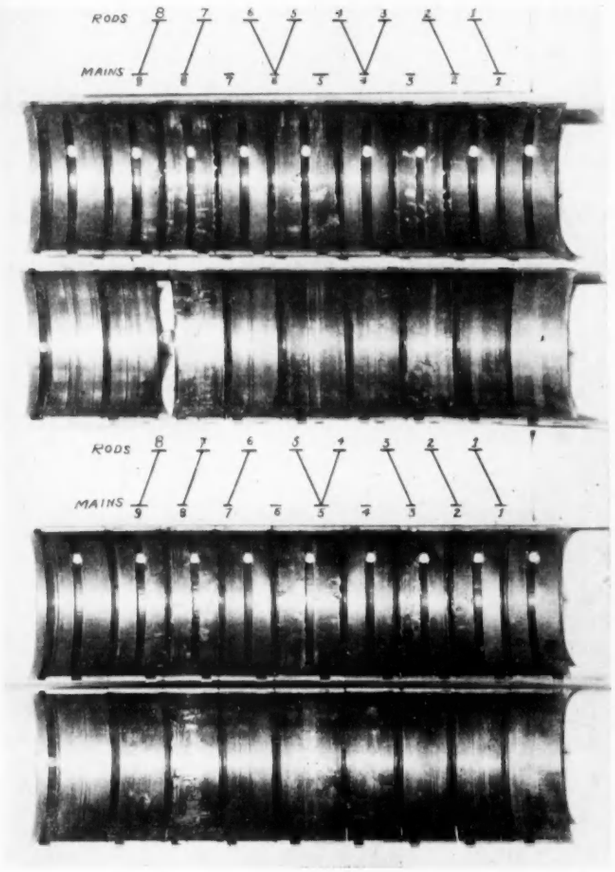


Fig. 4—Babbitt Main Bearings after 50 Hr. at 4500 R.P.M.

The poor condition of bearings 3, 5 and 7, in the upper group, is not due to the fact that they feed no rods. This is proved by the second test.

subjected to 50 hr. of full-load operation at 4500 r.p.m. Conditions were the same except for changes in the drilling of the crankshaft, such that bearings which supplied oil to two connecting rods in one test supplied oil to no rods in the other test. This change may have affected the temperatures of the bearings but it certainly had no major influence upon the damage incurred by the bearings during the test. A careful examination of Fig. 4 will show quite clearly that bearings 3, 5 and 7 have suffered because of crankshaft deflection rather than because of any peculiarity of oil flow.

Other Causes of Cracking.—Bearing failures which appear to result from faulty bonding may be due neither to this cause nor to flexing of the bearing shell. This was shown by experiments with the rods shown in Fig. 5. These rods were operated at 3800 r.p.m. and disaster was invited by undercutting to give a 50-per cent reduction in bearing surface. This produced an abundance of cracks but these do not extend into the undercut portion of the bearing. Hence, they are are not due to flexing of the rod or cap. Moreover, they are not due to faulty bonding for, in many cases, the cracks stop at the depth of the undercut and sections have started to shear along a line tangent to the undercut surface.

While the evidence clearly establishes the innocence of the two suspects, flexing and faulty bonding, it fails to point decisively to the real criminal. At the moment it seems most probable that the cracks are due to the tangential force between the journal and bearing at spots where the oil film is inadequate to prevent metal-to-metal contact. Fig. 6 contributes some evidence on this score. Grooving was for the sole purpose of accelerating the wear and identifying the regions where this wear is a maximum. The rods were operated for a few hours at speeds of 4000 and 4500 r.p.m. with an oil temperature of 300 deg. fahr., and an oil pressure below normal. Particular attention is directed to the portion of the bearing marked *A* in Fig. 6. Here, the bearing has worn to such an extent that the grooves are no longer visible and this is significant, because it is the location where cracks first appear.

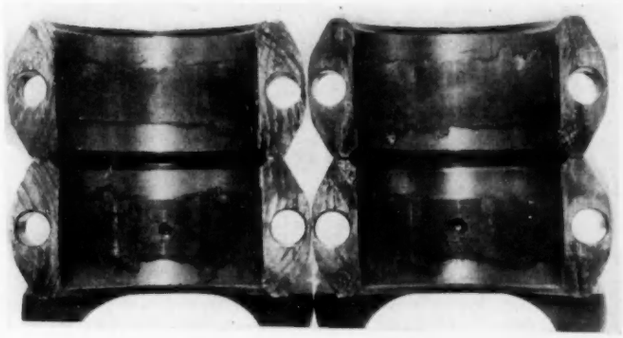


Fig. 5—Other Causes of Cracking

These bearings show that cracking is not necessarily due to flexing of the rod or faulty bonding. The cracks do not extend entirely through the babbitt and do not enter the undercut portion.

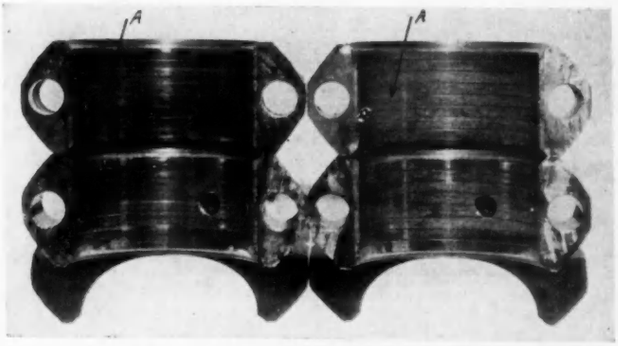


Fig. 6—The Bearings Shown Were Grooved To Show Areas of Maximum Wear

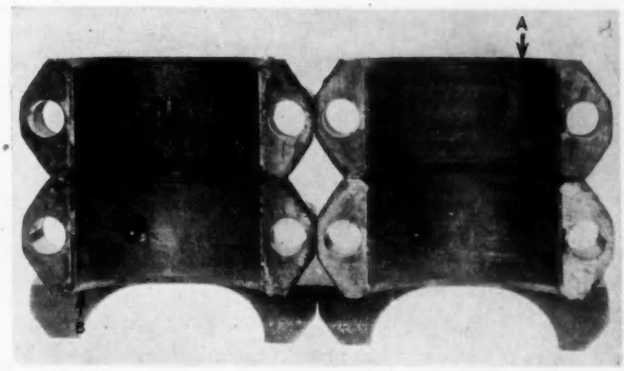


Fig. 7—The Bright Spots Show Metal-To-Metal Contact. Cracks Occur at the Trailing Edge, at *A*. Cracks at the Split, Due to Misalignment, Are Shown at *B*

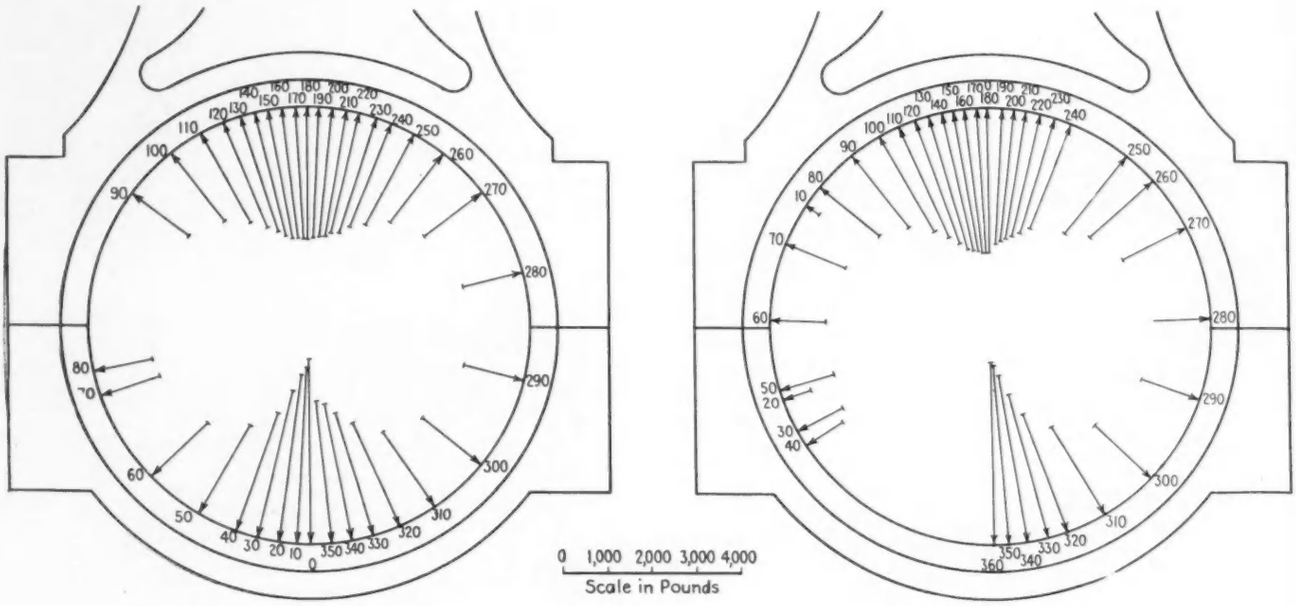


Fig. 8—Connecting-Rod Loads for Full Load and an Engine Speed of 4500 R.P.M.

The crankpin rotation is clockwise. At the left, the suction is 0 to 180 and the compression 180 to 360 deg. of crank angle past top center. At the right, the power is 0 to 180 and the exhaust 180 to 360 deg. of crank angle past top center.

The rod at the right in Fig. 7 furnishes further evidence along the same lines. This shows several bright spots indicating where the crankpin has broken through the film. At the trailing edge of each bright spot, a crack appears. As some of these cracks cannot be seen with the naked eye, it is fair to assume that the crack is a consequence of the metal-to-metal contact rather than that the bright spot is due to a displacement of metal brought about by the crack. The rod at the left in Fig. 7 merely shows cracking at the split because of a slight misalignment of the cap. That this misalignment exists is evident from the black deposit directly above the split.

Fig. 8 shows a conventional guess as to bearing loads resulting from gas pressure, inertia and centrifugal force. This does not indicate that the load on the bearing cap is a maximum in the region where the first cracks usually appear. It does show, however, that contact at this area occurs soon after the load on the rod shifts from tension to compression and it may be that there has been insufficient rotation at this time to provide a supporting oil film.

Thickness of Babbitt.—Modern production methods make it possible to use a thickness of babbitt of 0.015 in. or less. In our experience a thickness of less than 0.030 in. has been found undesirable. The thickness does not seem to have a major influence upon wear or cracking. With very thin babbitt, however, if a small piece breaks loose it can tip sufficiently to bring its edges in contact with the shaft. These are ground away rapidly and, eventually, the entire section disappears. With the thicker babbitt, there is not likely to be any loss in material until the cracks extend to the edge of the bearing. An example of this is shown in Fig. 9.

Undercutting To Extend Bearing Life.—In discussing the causes of cracking, it has been pointed out that cracks ordi-

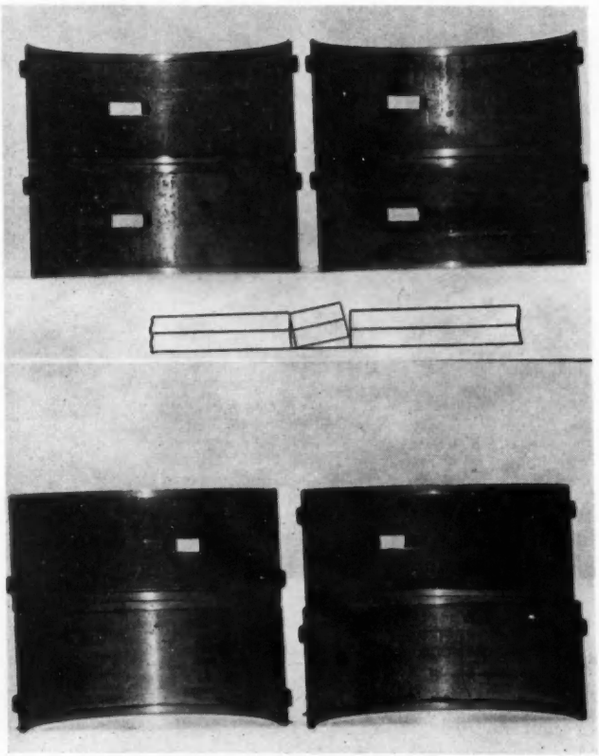


Fig. 9—Illustrating That Fragments of Thin Babbitt Escape More Readily from a Bearing Than with Thick Babbitt

The babbitt in the upper bearing is 0.010 in. thick; in the lower bearing, 0.034 in. thick.

narily do not extend into the undercut portion of the bearing. Fig. 10 shows how advantage may be taken of this fact to prolong bearing life. One of the two sets of bearings was undercut at the edges to a width of about 1/16 in. and a depth of 0.005 in. Both sets of bearings were subjected to the same test. The cracks are slightly more numerous in the bearings with the undercut edges, but the undercut portion has served as a barrier to prevent the cracks from spreading and permitting the babbitt to escape. As a result, although these bearings are marked by lines of "worry and care," they are still serviceable. Strangely enough, this is an instance where a bearing has been made better by making it worse, according to the tenets of those who worship at the shrine of the PV-factor or pay homage to the ZN/P-relationship.

Copper-Lead Bearings

To some extent copper-lead mixtures are superseding babbitt for main and connecting rod bearings. Such mixtures will be discussed primarily from the standpoint of their performance in comparison with that of babbitt. Matters of analysis and structure can best be left to those more intimately acquainted with the production of such bearings. It is generally understood, however, that this material has a melting point considerably above that of babbitt and that its physical characteristics are less affected by temperature. The material appears to crush as readily as babbitt and metal-to-metal contact produces a film of lead which presumably has sufficient lubricating value to prevent seizing. A typical analysis shows 45 per cent of lead, 2 per cent of nickel and the remainder is copper.

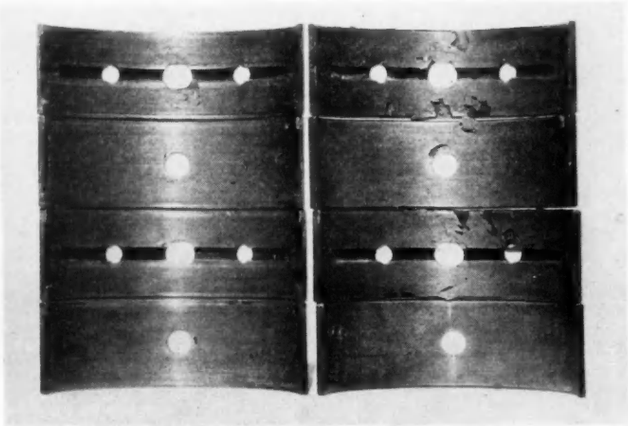


Fig. 10—Illustrating That Undercutting Bearings at the Edges Provides a Barrier Which Prevents Cracks from Spreading

The bearings at the left are undercut; those at the right are not undercut.

Life.—Figs. 11 and 12 show babbitt and copper-lead bearings after approximately 50 hr. of full-load operation at 4500 r.p.m. In several previous tests, failure of the babbitt bearings had made it necessary to discontinue the test after about 30 hr.

Crankpin Wear.—In early tests with the copper-lead mixtures, crankpin wear was excessive and much greater than with babbitt bearings. The excessive wear was eliminated by an increase in oil pressure and recent tests show no difference between babbitt and the copper-lead mixtures in this respect.

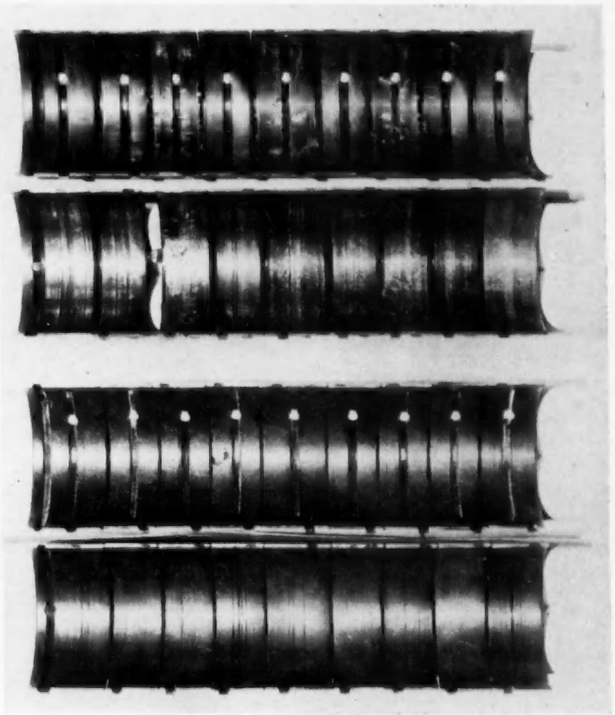


Fig. 11—Babbitt versus Copper-Lead Main-Bearings after 50 Hr. at 4500 R.P.M.

The upper bearings are of babbitt; the lower of copper-lead.

Overheating.—Too little clearance or an interruption of the oil supply causes babbitt to melt and the bearing is said to "burn out." In similar circumstances the copper-lead bearing wears rapidly. Apparently the lead "stews out" and furnishes

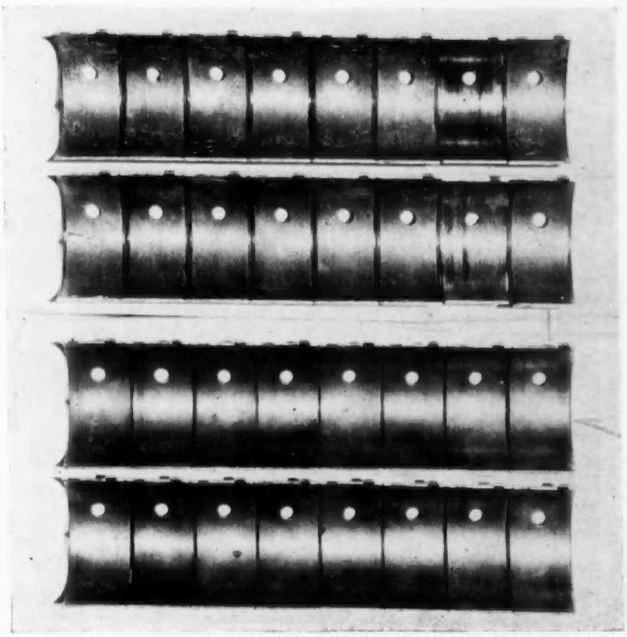


Fig. 12—Babbitt versus Copper-Lead Connecting-Rod Bearings after 50 Hr. at 4500 R.P.M.

The upper bearings are of babbitt; the lower of copper-lead.

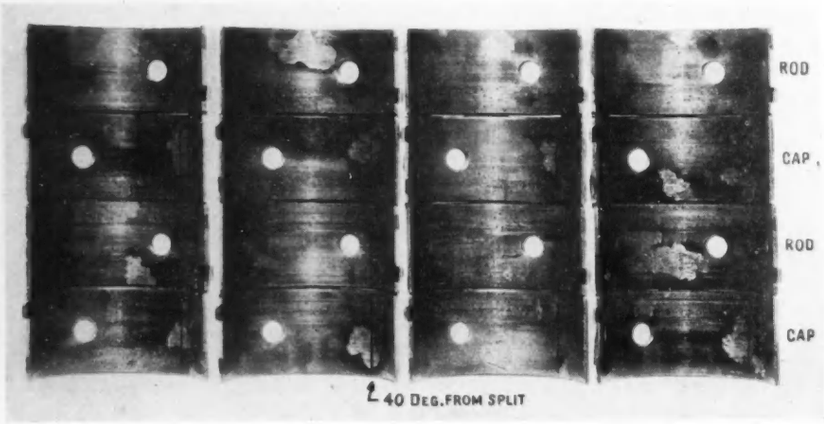


Fig. 13—Illustrating That "Blisters" Sometimes Form in Copper-Lead Bearings

The blisters project above the surrounding surface. Note that the blisters in the bearing caps occur in approximately the same location.

sufficient lubrication to prevent serious damage to the crankpin. Naturally, a failure of this type at high speeds does not benefit the crankpin, but results thus far indicate that less harm is done when the bearing is of copper-lead. Babbitt bearings replaced after such a failure usually have a short life. In a recent test, three copper-lead liners failed in succession before the reason for the lack of oil to this crankpin was discovered. When this was corrected, the fourth liner gave no trouble in the remaining 10 hr. at 4500 r.p.m.

Disintegration.—The type of failure which most nearly parallels the cracking of babbitt consists in the crumbling and washing away of portions of the copper-lead. Prior to such failures, bright spots appear as indicated in Fig. 13. Measurements show that these spots are higher than the surrounding surface. Frequently, such spots do no particular damage and after the bearing is cleaned it appears to be in good condition. Eventually, however, disintegration is likely to start at these spots. At present it is not possible to state positively whether this condition is due to faulty structure or to the presence of foreign material in the bearing. When this trouble appears it is most likely to be found in those areas where it is difficult to maintain an oil film. In Fig. 13, for example, it will be seen that the spots in the bearing caps are about 40 deg. from the split in the direction of crankpin rotation. The rise in temperature which accompanies metal-to-metal contact causes expansion and, if the surrounding cooler metal prevents expansion along the surface, the natural result will be for the spot to bulge or "blister." This will still further increase the tendency to break through the oil film.

Unsuitable Oil.—Certain oils have proved to be unsuitable for copper-lead bearings. The action of such an oil is illustrated in Fig. 14, which shows two sets of bearings, one of which was operated with a suitable oil and one with an unsuitable oil. The test gives only light loads and consists in spinning a crankshaft for 20 hr. at 1500 r.p.m. With the unsuitable oil, chemical action takes place with the lead and the resulting product is washed away, leaving a porous copper structure. With an engine in normal operation, the porous structure wears away very rapidly. With the bearings shown in Fig. 15 there was an increase in clearance of more than 0.003 in. in less than 10 hr. of operation at 4500 r.p.m.

With respect to this situation, several facts should be made clear. In the first place, the unsuitable oils are not bad oils but are frequently oils which have a high degree of "oiliness" and have hitherto given excellent results. In the second place, the inability of these bearings to give satisfactory performance with certain oils should be recognized as a definite disadvantage. On the other hand, it should not be forgotten that engines are already fairly fussy about the viscosity of the oil they use and about the characteristics of the fuel they burn. The important question is whether the providing of a satisfactory oil for these bearings will impose undue hardship on the oil producer or car user. This seems improbable, as the majority of oils on the market are entirely satisfactory. Incidentally, both of the oils used in the tests shown in Fig. 14 were furnished by the same producer and it is understood that the unsuitable oil has been withdrawn from the market. All

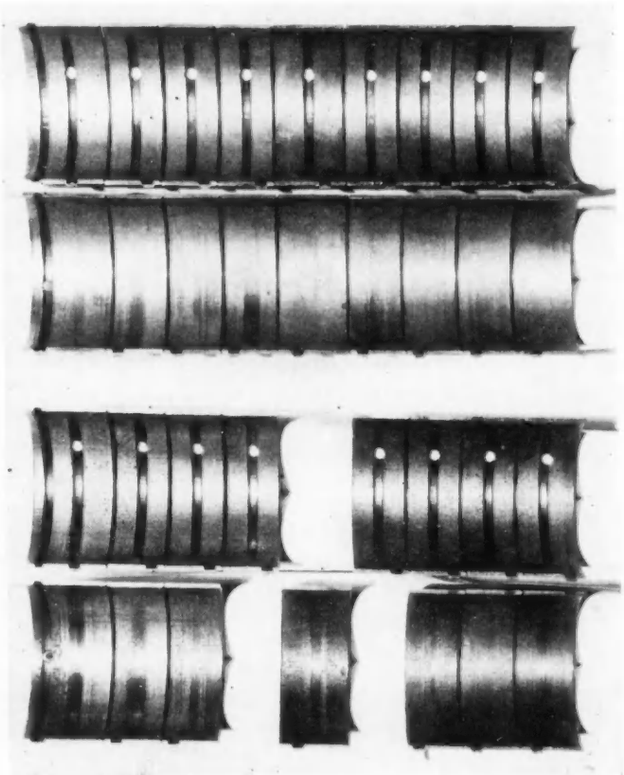


Fig. 14—Spinning Test for Copper-Lead Bearings

A crankshaft was spun in the bearings shown for 20 hr. at 1500 r.p.m., using a suitable oil for the upper set and an unsuitable oil for the lower set. With an unsuitable oil, the lead is washed out, leaving a porous structure.

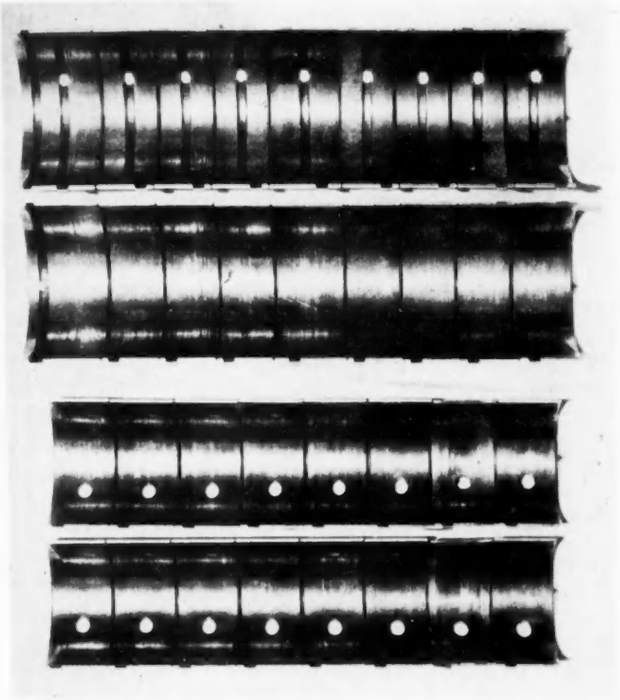


Fig. 15—Copper-Lead Bearings after 10 Hr. at 4500 R.P.M.

The clearance increased more than 0.003 in. because of the unsuitable oil used.

of this discussion may be superfluous, as Fig. 16 shows that some of the recent copper-lead mixtures are but little affected by these special oils.

Bearing Shells.—A few years ago main-bearing shells were usually of bronze, with a thickness of about 3/16 in. Later, the use of steel in place of bronze became common and, more recently, the thin-back bearings with steel shells of about 1/16 in. thickness have found favor. With connecting rods, the choice is usually between the thin-back liners and direct bonding of the bearing material to the rod. The selection has been dictated largely by considerations of cost, uniformity of bonding, ease of replacement and the like. In the tests with which I am familiar, all types have given essentially the same life, provided the bonding was equally good. In installations where bearing life is sensitive to slight changes in the rate at which heat is conducted from the bearing to the case, the story might be different. When it was found that no perceptible difference resulted from grooving the shell so as to reduce the contact area by 50 per cent, further worry on that score did not seem justified.

Mention was made earlier in the paper of experiments which showed that the "crush" of thin-back bearings had no pronounced effect upon the tendency of the bearings to crack. In our experience, the best results have been obtained with limits such as to give minimum crush without permitting a lack of crush under any conditions. In other words, the total circumference of the bearing shells should never be less than the circumference of the hole in the case. Too much crush puts unnecessary strain on the bearing bolts and tends to distort the hole in the case. If a bearing seizes, too little crush may permit it to break its retaining lugs and turn with the shaft.

Lubrication.—Comments on lubrication are intended merely to point out the possibility of bearing failures at high speeds because of the increased demand made upon the lubricating system. Fig. 17, for example, shows the amount of oil necessary to maintain a pressure of 40 lb. per sq. in. This increases rapidly with increase in number of revolutions per minute, since the discharge is governed largely by the centrifugal force of the oil in the crankpins and this varies with the square of the speed. The upper curve shows the effect of removing the rods and permitting unrestricted discharge from the ¼-in. holes in the crankpins. In this case the discharge at high speeds is much less than at low speeds. In the dark days of depression, the master minds of the economic realm discovered that the public was spending less money because it was receiving less. The same explanation applies to the upper curve in Fig. 17, and it is highly probable that at each speed the crankpin could discharge more oil if it received more.

As engines rarely operate without connecting rods, this becomes of importance only when the two curves are seen to cross at 4500 r.p.m. This crossing suggests that, with the engine operating normally, the crankpins are discharging all the oil received and that the lubrication of the rod is limited by the ability to get oil into the crankpin. This belief is supported by Fig. 18, which shows the effect of changing the length of the grooves in the main bearings. At 4500 r.p.m., practically no oil enters the crankpin when the groove in the main bearing is replaced by a single hole. As the surface of the main journal is traveling at more than 30 m.p.h. it is not surprising that oil, after slumbering with the dinosaurs

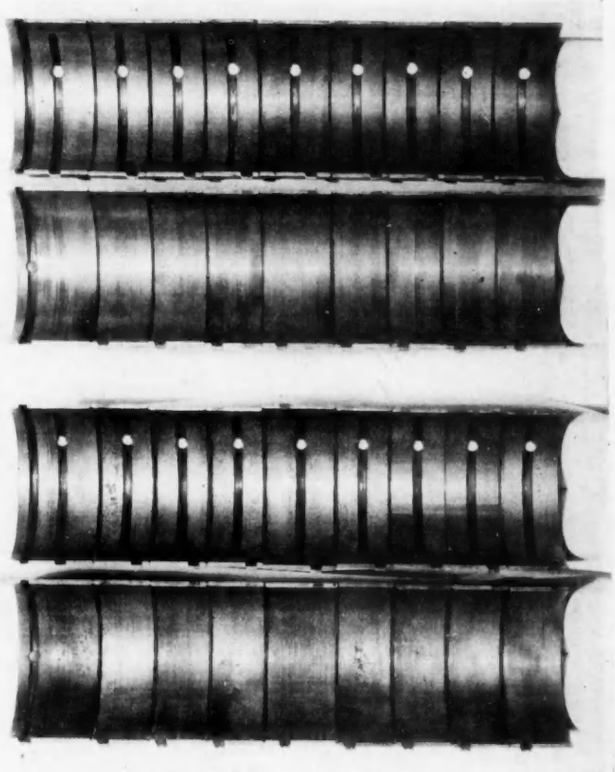


Fig. 16—Copper-Lead-Mixture Bearings after a Spinning Test

Recently developed copper-lead mixtures are little affected by unsuitable oils. The upper set of bearings is composed of the new mixture; the lower, of the original mixture.

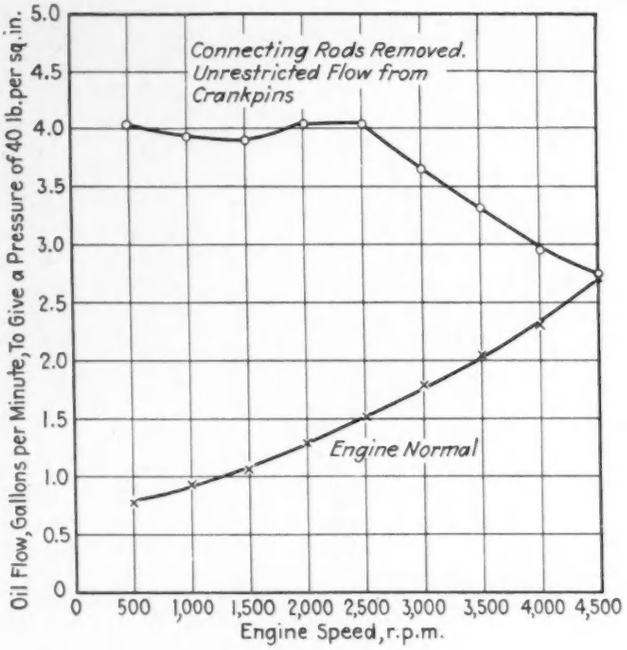

Connecting Rods Removed. Unrestricted Flow from Crankpins
Engine Normal
Oil Flow, Gallons per Minute, To Give a Pressure of 40 lb. per sq. in.
Engine Speed, r.p.m.
0 0.5 1.0 1.5 2.0 2.5 3.0 3.5 4.0 4.5 5.0
0 500 1,000 1,500 2,000 2,500 3,000 3,500 4,000 4,500

Fig. 17—Amount of Oil Required To Maintain a Pressure of 40 Lb. per Sq. In. in the Main Oil-Line, the Oil Viscosity Being 108 Saybolt Sec. at 150 Deg. Fahr.

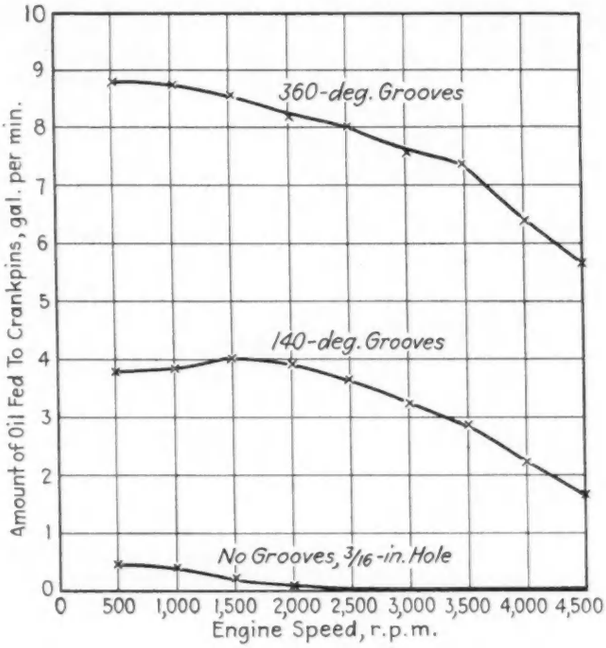

360-deg. Grooves
140-deg. Grooves
No Grooves, 3/16-in. Hole
Amount of Oil Fed To Crankpins, gal. per min.
Engine Speed, r.p.m.
0 1 2 3 4 5 6 7 8 9 10
0 500 1,000 1,500 2,000 2,500 3,000 3,500 4,000 4,500

Fig. 18—Curves Showing Oil Flow versus Length of Main-Bearing Grooves

The oil pressure was 40 lb. per sq. in.; the oil temperature, 150 deg. fahr.; and the oil viscosity at that temperature, 108 Saybolt sec.

for a few million years, should lack the agility to leap aboard. This single example will suffice to show that lubrication of the connecting rod involves more than drilling a couple of holes and cutting some sort of a groove. It is hardly necessary to point out that too little oil supplied to the crankpin means lubrication of the feast-and-famine type, with a large share of the oil discharged during that portion of the revolution immediately following registry.

Leakage Area.—As lubrication is dependent upon maintaining a certain pressure in the main oil-line, it is pertinent to consider factors which affect oil pressure. Leakage areas have been calculated for one engine with nine main bearings grooved for 140 deg. Assuming that the shaft is central and that the diametral clearance is 0.001 in., the leakage area is found to be 0.026 sq. in., which is slightly greater than the area of two ⅛-in. holes. If the clearance is increased to 0.002 in., either by leakage or initial fit, the leakage area is doubled. The significance of this will be apparent from Fig. 19, which

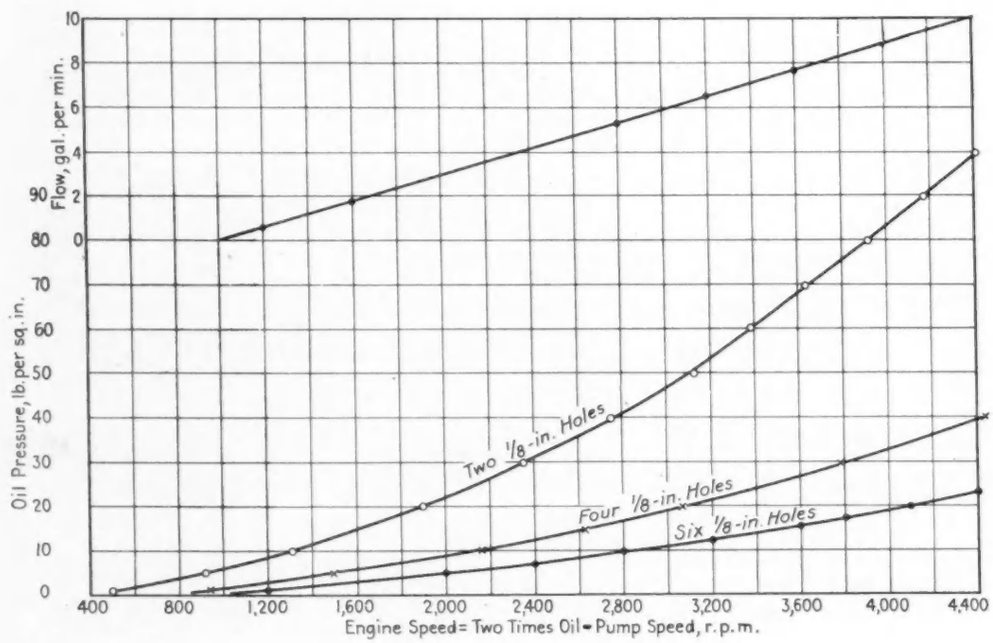

Two 1/8-in. Holes
Four 1/8-in. Holes
Six 1/8-in. Holes
Oil Pressure, lb. per sq. in.
Flow, gal. per min.
Engine Speed = Two Times Oil-Pump Speed, r.p.m.
0 10 20 30 40 50 60 70 80 90
0 2 4 6 8 10
400 800 1,200 1,600 2,000 2,400 2,800 3,200 3,600 4,000 4,400

Fig. 19—Curves Showing Oil-Pump Characteristics

The oil temperature was 180 deg. fahr.; the oil viscosity at that temperature, 70 Saybolt sec.

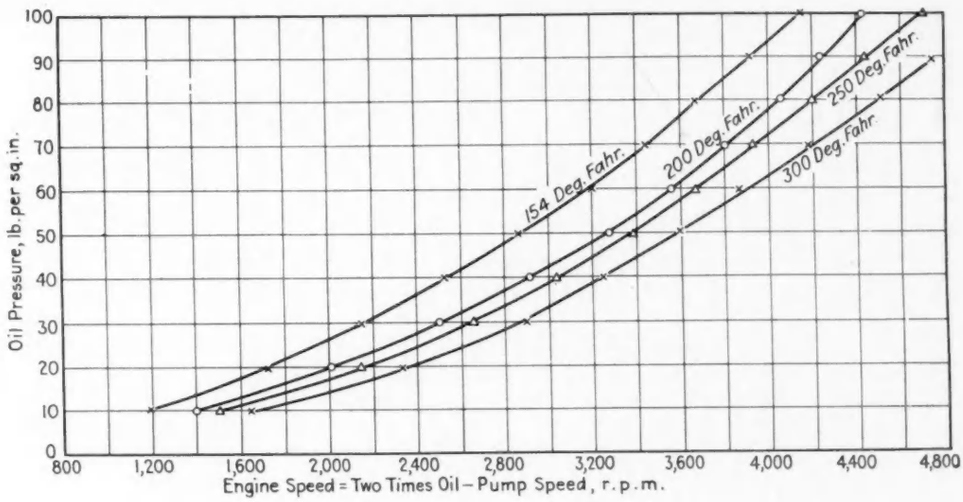


Fig. 20—Curves Showing the Effect of Temperature on Oil Pressure

The oil viscosity at 150 deg. fahr. was 108 and, at 250 deg. fahr., 43 Saybolt sec.

shows the pressures which can be maintained by a given oil pump when discharging through two, four and six ⅛-in. holes.

Temperature.—Fig. 20 shows pressure curves of the same pump with a constant discharge-area and various oil-temperatures. With the viscosity lowered by an increase in temperature, more oil will be fed to the crankpin for a given main-line pressure; but, by the same token, the oil will be discharged from the crankpin at a more rapid rate. The end leakage at the main bearings is increased and, if the oil pump is working to capacity with cold oil, an increase in temperature increases the likelihood that the rod bearings will not receive adequate lubrication throughout the entire revolution. Moreover, at extremely high temperatures, the viscosity of the oil may not be adequate to carry the load.

There is room for argument as to the value of an oil cooler on an engine where the temperature of the oil never rises higher than that of the jacket water. There is not much question, however, as to the desirability of preventing oil temperatures from going much above 200 deg. fahr. Special provision for oil cooling is fast becoming a necessity, partly because of higher engine-speeds and partly because of the tendency to tuck engines away where they receive very little cooling from the outside air. One example of the effect of speed is furnished by a recent engine test in which cooling, which was sufficient to give a temperature of 100 deg. fahr. at 1000 r.p.m., gave 117 deg. at 2000, 142 deg. at 3000, 169 deg. at 4000 and 196 deg., at 4500 r.p.m.

Dirt.—Fig. 21 shows the effect of feeding sand to an engine in which only half of the bearings are protected by a filter. This illustration may be inappropriate, as there are few engines which are not provided with filters as well as with means for preventing dirt from entering the engine. It will serve, nevertheless, to call attention to the centrifuging action of the crank which causes the dirt to be discharged at the edge of the oil hole. It is appropriate to call attention to the fact that, with the high rate of oil flow necessitated by high speed, there is an increased tendency for the dirt in the oil pan to be swept up and carried to the filter. This necessitates greater filtering capacity. Even with carefully filtered oil, scratched bearings are not uncommon. Lightning may not strike twice in the same place; but it usually does sufficient damage the first time, and a small particle of bearing material can do plenty of harm in a single trip through a bearing. With inadequate filtering, scratches are usually found on those bearings which, even with clean oil, show maximum wear. Hence, sufficient oil of proper viscosity offers considerable protection against damage from dirt which cannot be prevented from reaching the bearings.

Conclusion.—It is realized that this discussion is incomplete, and many items of importance have been omitted in the belief that their influence is well understood. Kipling once said:

> "——What avails the classic bent
> And what the cultured word,
> Against the undoctored incident
> That actually occurred?"

The sole object of this paper has been to describe a few such undoctored incidents in the hope of throwing light on bearing problems.

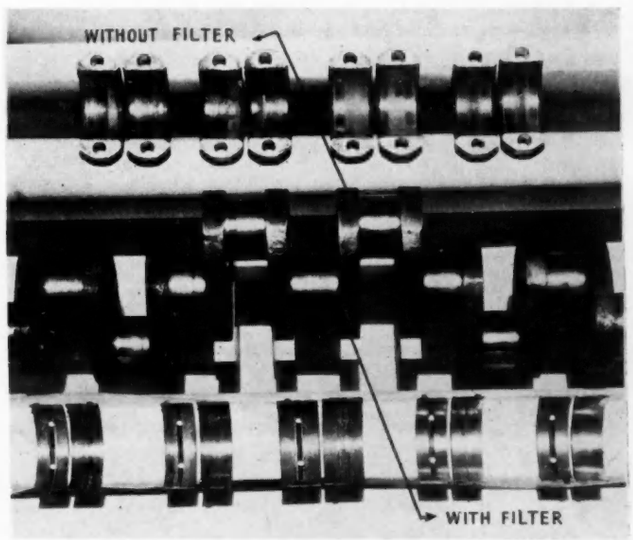


Fig. 21—Result of Dirt Test of Bearings, both With and Without an Oil Filter

The upper bearings were lubricated without using a filter; the lower, with a filter. Note how wear is confined to one end of the crankpin.

Winter Oils for Automobile Engines

By W. H. Graves[1], H. C. Mougey[2], and E. W. Upham[3]

THE Society of Automotive Engineers has always appreciated the close relationship between fuels, lubricants, automotive design and operating conditions. In the very early days, low-viscosity engine-oils were regularly used; but, as crankcase dilution began to be recognized and as the effect of viscosity on oil consumption began to be more important, the use of oil of higher viscosity became more general and has continued to the present time to the great inconvenience of motorists, especially in cold weather.

Electric starters were introduced in 1911. In 1913 the Metropolitan Section of the Society held a symposium on starters. At this meeting C. E. Wilson[4] presented curves showing the torque required to crank engines, both warm and cold, and the torque available from the starter and battery. He pointed out that in cold weather the torque required is greatly increased due to the thickened oil. In a spirited discussion[5] of the papers at this symposium, it was brought out that a minimum cranking speed was necessary in order to get a combustible mixture distributed to the cylinders and numerical values from about 40 r.p.m. up to 100 r.p.m. seemed to be favored.

Stratford[6], in 1915, gave data showing that higher-viscosity oils give more miles per gallon of oil, but at the cost of decreased power, less acceleration and greater fuel consumption. He also gave curves showing the change in viscosity with change in temperature with oils of different types. Kettering[7], in 1916, in a paper on the control of temperature in engines, pointed out that the thickened lubricating oil at low temperatures very greatly increased the difficulty in cranking engines fast enough to get the gasoline mixture distributed to the cylinders, and A. Ludlow Clayden[8], in discussing the paper, predicted that it was "absolutely essential" to do something to improve winter starting or trouble would continue to be experienced.

Apparently all the individual factors of oil and gasoline consumption, cranking speed, and the effect of temperature on oil viscosity and engine friction, were known at that time, and even the importance of doing something to correct these winter starting troubles was recognized. However, it seems that the necessary facts were not generally enough known or appreciated, since, as predicted by Clayden, winter starting troubles still continued.

MacCoull[9], in 1921, published his viscosity-temperature chart, by which it was possible to estimate the viscosity of oil at any temperature if its viscosity at two other temperatures was known. Herschel[10], in 1922, gave a paper on Change in Viscosity of Oils with Temperature, followed a few years later by his viscosity-temperature chart[11]. Several other charts by various authors appeared in rapid succession, each attempting to express more accurately the relationship of viscosity to temperature.

In 1923, Clayden[12], discussing crankcase-oil dilution, pointed out that it is dirt and water rather than the gasoline in the oil that causes the real trouble. He said: "Oil which is free from dirt will give safe lubrication even when very considerably reduced in viscosity." He also said: "From the lubrication viewpoint there are three principal things to consider; (*a*) the viscosity of the oil under piston temperatures, (*b*) the viscosity of the oil at starting temperatures, and (*c*) the behavior of the oil when subjected to combustion-chamber temperatures." In 1923, and for several years previous, the Cadillac oil specifications were as shown in Table 1.

In an attempt to offset extremely heavy dilution, some of the oil companies wished to supply oils of higher viscosity,

Table 1—Cadillac 1923 Oil Specifications

	Summer, Above 32 Deg. Fahr.	Winter, Below 32 Deg. Fahr.
Cold Test (Maximum), deg. fahr.	20	0
Flash Point (Minimum), deg. fahr.	380	380
Saybolt Viscosity at 100 deg. fahr.	300-450	170-200
Conradson Carbon (Maximum), per cent	0.3	0.2

Table 2—Viscosities of Oils Used in Vacuum Oil Co. Tests

	Saybolt Viscosity of Oils as Shown in Report[13] of March, 1923			Saybolt Viscosity of Oils of Same Trade Names[a], as Marketed in 1934		
	At Temperatures of, Deg. Fahr.					
	210	100	0	210	100	0[b]
Mobiloil "A"	58	585	270,000	65-66	600	95,000
Mobiloil "Arctic"	49	285	45,000	58	380	31,000
Mobiloil "Arctic" + 5-Per Cent Dilution	..	238	12,000	....	...	....

[a] From a private communication to the authors.
[b] Viscosity obtained by using A.S.T.M. chart D341-32T.

[This paper was presented at the Semi-Annual Meeting of the Society, Saranac Inn, N. Y., June, 1934.]

[1] Chief Metallurgist, Packard Motor Car Co., Detroit. Chairman of the Publicity Subcommittee of the S.A.E. Lubricants Division.

[2] Assistant Technical Director, Research Division, General Motors Corp., Detroit. Vice-Chairman of the S.A.E. Lubricants Division.

[3] Chief Metallurgist, Chrysler Corp., Detroit. Chairman of the S.A.E. Lubricants Division.

[4] See S.A.E. BULLETIN, vol. 4, pp. 308-313, June, 1913; Desirable Characteristics of Starting Motors, by C. E. Wilson.

[5] See S.A.E. BULLETIN, vol. 4, pp. 359-363, July, 1913; Discussion at the June, 1913, meeting of the Metropolitan Section.

[6] See *Transactions of the Society of Automobile Engineers*, vol. 10, part 2, pp. 86-109, 1915; Automobile Lubrication, by C. W. Stratford.

[7] See S.A.E. BULLETIN, vol. 10, pp. 685-694, September, 1916; Engine-Temperature Control, by C. F. Kettering.

[8] See S.A.E. BULLETIN, vol. 10, pp. 696-698, September, 1916; Operation on Heavy Fuels, by A. Ludlow Clayden, in discussing the paper on Engine-Temperature Control, by C. F. Kettering.

[9] See *Lubrication*, vol. 7, pp. 5-8, June, 1921, published by The Texas Co.; A New Chart for Viscosity-Temperature Relations, by Neil MacCoull.

[10] See *Journal of Industrial and Engineering Chemistry*, vol. 14, pp. 715-723, August, 1922; Change in Viscosity of Oils with Temperature, by Winslow H. Herschel.

[11] See *Oil and Gas Journal*, vol. 25, pp. 146, 150, 176, 178, 182 and 184, Dec. 2, 1926; Viscosity and Temperature Changes, by Winslow H. Herschel.

[12] See *Automotive Industries*, vol. 49, pp. 415-417, Aug. 30, 1923; Dilution of Crankcase Oil Is Serious Factor Even in Summer, by A. Ludlow Clayden.

THE factors involved in cold starting of automobile engines, including the effects of temperature and oil viscosity on cranking speed and torque, have been known for many years. Many papers have been presented before the various Sections of the Society on these subjects.

The S.A.E. crankcase-oil viscosity-numbers, which were adopted in July, 1926, provided for the classification of the lower-viscosity oils at 130 deg. fahr. and the higher-viscosity oils at 210 deg. fahr. It was recognized by 1930 that a classification for winter oils must be based on the viscosity of the oil at the starting temperature, and work was started on this problem. In June, 1933, the 10-W and 20-W oils, which are classified in accord with their viscosity at 0 deg. fahr., were adopted for publication and trial.

The results of the use of these oils during the winter of 1933-1934, together with their advantages, are discussed. The data show that these oils are necessary for cold starting, provide adequate lubrication and improved performance and give reasonable oil mileage.

Although higher-viscosity oils give more miles per gallon of oil, by far the most important factor in oil consumption is engine speed. The lower-viscosity oils, that are required for winter operation, reduce engine friction; and the total cost of oil plus gasoline, using oils of lower viscosity, may be less than when using oils of higher viscosity.

especially in winter. In order to determine definitely the grade of oil best suited for winter use in engines of this type, the Vacuum Oil Co.[13], in March, 1923, made a very elaborate series of tests in its cold room, using a Cadillac Type-61 (V-8) engine. Copies of the report of these tests, although never published, were given by the Vacuum Oil Co. to interested parties including Cadillac and Delco engineers. This report deals with practically all of the factors which are recognized at the present time as affecting winter starting. Data and discussion are given in regard to such factors as Saybolt viscosities of fresh and diluted oils over the entire temperature range of from 210 deg. fahr., to 0 deg. fahr., torque required to crank at various speeds using the various oils over the temperature range of from 30 deg. fahr. to 0 deg. fahr., cranking speed using both warm and cold batteries, minimum cranking speed necessary to permit starting the engine, pumpability of the oil at low temperatures, and the like. The viscosities of the oils used in this test, as shown in the report, are given in Table 2. Of course, these 1923 values do not apply to the present oils of these trade names. To avoid any possible misunderstanding as a result of quoting these 1923 tests, data, and conclusions, the 1934 Saybolt viscosities of the oils of these same trade names are also shown in Table 2.

The last paragraph of the "conclusions" from this report, together with the "recommendations," are as follows:

"Where conditions are such that the oil and battery both reach low atmospheric-temperatures, due to lengthy exposure in zero weather, starting with Gargoyle Mobiloil 'A' will be impossible unless the oil is quite badly diluted. With Gargoyle Mobiloil 'Arctic,' conditions can be much more severe without occasioning starting difficulties.

"Recommendations.—The results shown by this test clearly indicate the superiority of Gargoyle Mobiloil Arctic to Gargoyle Mobiloil 'A' as a winter lubricant for the Cadillac Model-61 engine as far as starting conditions are concerned.

"Knowing that Gargoyle Mobiloil 'Arctic' will provide satisfactory lubrication and protection to the engine and that it will be positively circulated by the oil pump at zero temperatures, the winter recommendation on the Cadillac Model-61 engine has been made Gargoyle Mobiloil 'Arctic.'"

Attention is called to this record of tieing up the Saybolt viscosity at 0 deg. fahr. with the oil recommended for starting and for regular winter use.

At the Chicago Section Meeting of the Society in November, 1924, J. H. Hunt and O. C. Berry both presented papers on Cold Starting[14]. They both pointed out that crankcase dilution, about which engineers had been worrying for years, was really not so bad after all. The report of this meeting in the S.A.E. JOURNAL says: "J. H. Hunt agreed with Mr. Berry that a certain degree of dilution is desirable in winter, stating that starters of conventional capacity cannot function satisfactorily at low temperatures unless the so-called winter oils are diluted about 15 per cent. Starting motors of larger torque-capacity can be built by the electrical engineer, but the motor-car manufacturers do not appear willing to pay the increased cost that this would entail. Mr. Hunt said that it is his practice to dilute fresh oil with kerosene before putting it in engines in a Cold Room."

The Bureau of Standards[15], in January, 1925, reported on work it had been doing on pressure-viscosity, and showed that both pour-point and viscosity at low temperatures are important in determining the performance of oils at low temperatures. Previously, there had been a general impression in the minds of the public that the suitability of an oil for low-temperature operation depended upon the pour-point, without giving much consideration in addition to the viscosity of the oil at low temperatures. In fact, it was erroneously believed by many that oils actually turned solid and that the viscosity became infinite when the pour-point was reached. Comparatively few people realized that some low pour-point oils were too viscous at low tempeartures for winter use, and that some oils having pour-points above 0 deg. fahr. were actually low in viscosity, from the standpoint of resistance to engine cranking at low temperatures, and that oils might flow to the pump inlet at temperatures several degrees below the pour-point. This confusion on the subject of the importance of both pour-point and of viscosity at low temperatures seriously interfered with the selection and use of the oils

[13] See Report on Low-Temperature Starting-Test on Cadillac Model-61 Engine, being a confidential report made in March, 1923, by the Engineering Division of the Vacuum Oil Co. to the Cadillac Motor Car Co.

[14] See S.A.E. JOURNAL, vol. 15, pp. 488-489, December, 1924; Dilution Needed for Easy Starting.

[15] See S.A.E. JOURNAL, vol. 16, pp. 116-118, February, 1925; Lubrication and Dilution.

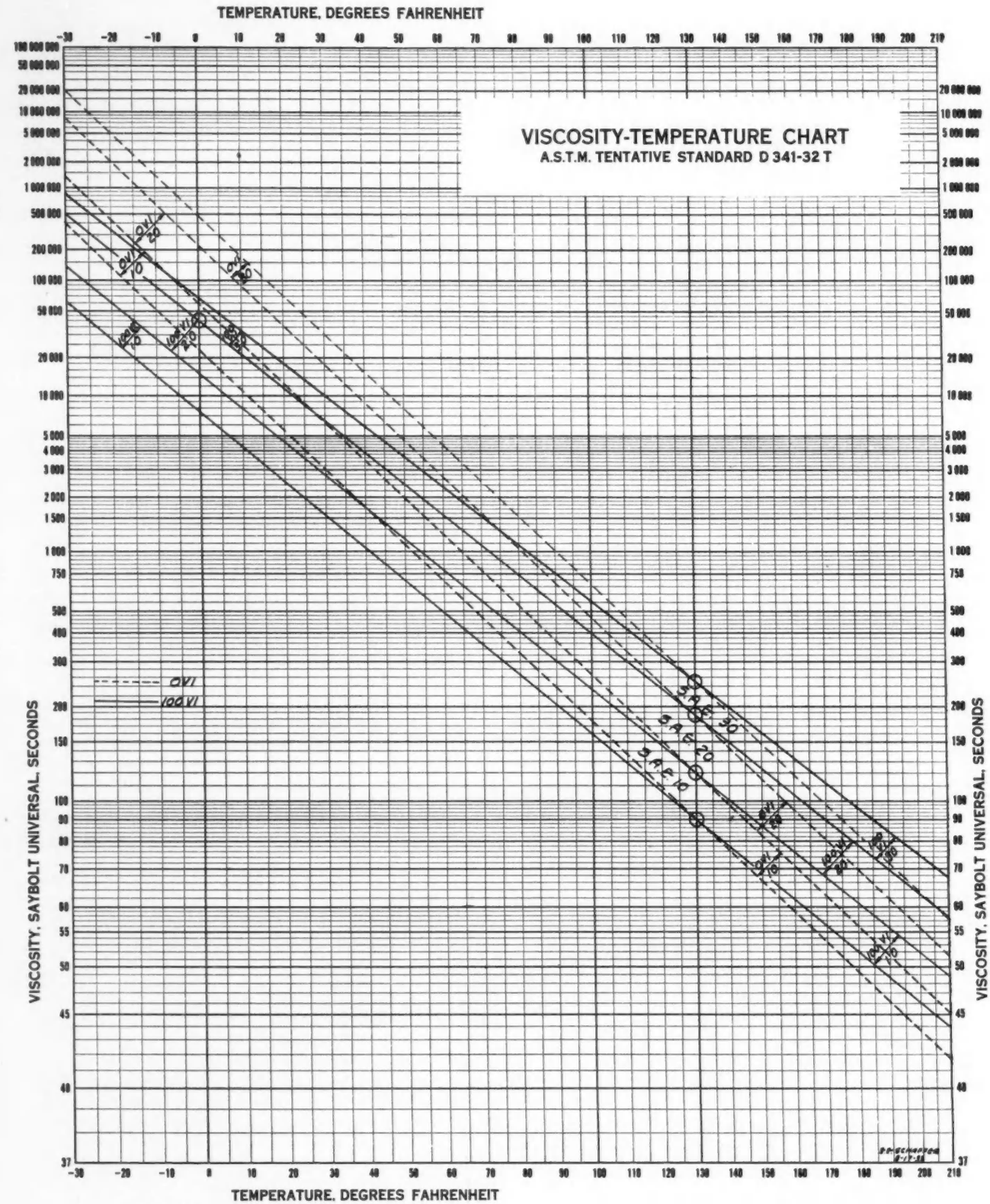


Fig. 1—Differences between 10-W and 20-W Oils and the Classifications S.A.E.-10, S.A.E.-20 and S.A.E.-30 Oils

best suited for winter service, although the facts were known and in many other papers, in addition to the ones mentioned, the various factors had been discussed.

It should be noted that by January, 1925, practically all the important factors on cold starting and winter lubrication had been developed and published, but many of these factors were hid in a mass of other detail and discussion. However, at the annual meeting of the Society in Detroit, January, 1928, Wilkin, Oak, and Barnard[16] took all these separate building blocks, added to them very greatly from their own work, and presented as a complete structure their classical paper on Motor Oil Characteristics and Performance at Low Temperatures. They concluded that ". . . . . the data indicate consistently that the cranking efforts observed were determined by oil viscosity . . . . . at starting temperature, calculated from viscosity-temperature curves." They gave data that showed just what viscosity in Saybolt seconds the winter oils for different engines should have.

The S.A.E. Lubricants Division had been working on the problem of S.A.E. viscosity numbers for crankcase oils for a long time and at the July, 1926, meeting of the Society, the S.A.E. viscosity numbers, in practically their present form, were adopted. With all the information that was available, it seems hard to explain at this time why the Lubricants Division did not hold a meeting in 1928 following the presentation of the paper by Wilkin, Oak and Barnard, and develop classifications for oils suitable for winter use; but the fact remains that it was not until May, 1930, that the Lubricants Division decided that a new classification for oils necessary for cold starting was desirable and "it was voted that the Chairman should appoint a Committee to investigate the subject of a 'cranking test' or 50,000 viscosity-temperature test." This Committee, in addition to the commercial difficulties which were to be expected, found that it was greatly handicapped by the fact that there were several different viscosity-temperature charts in general use. Although these charts were very similar, the numerical values obtained with the various charts were slightly different, and before any classification system could be proposed, a standard chart that could be used universally had to be developed. An appeal was made to the American Society for Testing Materials and, in 1932, the A.S.T.M. viscosity-temperature chart[17] was adopted.

While this work was going on, the subject of cold starting and winter oils was receiving a large amount of attention. Three outstanding papers were presented on this subject in 1931. Blackwood and Rickles[18] of the Standard Oil Development Co., Larson[19] of the Sinclair Refining Co., and Kent[20] of the Chrysler Corp. gave papers in which they specified the minimum cranking speed and the maximum viscosity at 0 deg. fahr. for winter oils. The data given in two of these papers make it possible to recalculate the viscosities in terms of the A.S.T.M. chart, although they report the viscosity values in their papers according to different standards. Their conclusions are shown in Table 3.

The work which has just been reported was duplicated and checked in many other laboratories and in service, and as a result, the S.A.E. Lubricants Division in June, 1933, approved for "information and trial" the two winter-oil classifications 10-W[21] and 20-W[22].

The difference between 10-W and 20-W and the classifications S.A.E. 10, S.A.E. 20, and S.A.E. 30, is shown in the charts in Figs. 1 and 2 and in Table 4. S.A.E. 10, S.A.E. 20 and S.A.E. 30 are classified in accord with their viscosity at 130 deg. fahr., while 10-W and 20-W are classified in accord with their viscosity at 0 deg. fahr., a temperature more closely associated with starting problems.

Since oils from different crudes have different rates of change in viscosity with change of temperature, and since each classification covers a range of viscosities, it is evident that oils whose viscosities are in a certain class at summer temperatures may be quite different when the thermometer is hovering around zero. The heaviest oil that can be classified as S.A.E. 10 may be 60,000 at 0 deg. fahr., while the lightest S.A.E. 30 oil may be only 40,000 at the same temperature. In other words, it might be possible for a car owner who knew that he should use oils of lower viscosity in winter to drain his summer oil, S.A.E. 30, and replace it with S.A.E. 10 that, instead of being lighter at winter temperatures, might be one and one-half times as heavy, and the heaviest S.A.E. 20 might be 225,000 at 0 deg. fahr., or almost six times as heavy as the lightest S.A.E. 30. By classifying oils for winter use in accord with their viscosities at winter temperatures, this confusion is avoided; and oils which will permit winter starting and operation may be obtained by the car owner.

On account of commercial conditions most 10-W oils, regardless of the crude from which they are made, will be approximately 8000 at 0 deg. fahr. and most 20-W oils, regardless of the crude, will be about 30,000 at 0 deg. fahr.

Table 3—Conclusions Regarding Viscosity

Investigators	Maximum Cranking Speed, R.P.M.	Maximum Viscosity for Starting at 0 Deg. Fahr.	
		Values Stated in the Paper, Sec.	Values Calculated in Terms of A.S.T.M. Chart, Sec.
Blackwood and Rickles	20 to 40	18,000 to 40,000 from Standard Oil Development Co. Chart	12,000 to 26,000
Larson	35	Oil on the Cylinder walls, 30,000. Undiluted oil, 50,000. From Sinclair Refining Co. Chart	Oil on the Cylinder Walls, 18,000. Undiluted Oil, 30,000
Kent	40	Standard Winter-Test Oil with 15 Per Cent of Kerosene Dilution	

Table 4—Differences between Classifications

Oil	Saybolt Viscosity, Sec. At 0 Deg. Fahr. Minimum	At 0 Deg. Fahr. Maximum	At 130 Deg. Fahr. Minimum	At 130 Deg. Fahr. Maximum, Less Than:
S.A.E. 10	.....	.....	90	120
S.A.E. 20	.....	.....	120	185
S.A.E. 30	.....	.....	185	255
10-W	5,000	10,000	...	...
20-W	10,000	40,000	...	...

[16] See S.A.E. Journal, vol. 22, pp. 213-220, February, 1928; Motor-Oil Characteristics and Performance at Low Temperatures, by R. E. Wilkin, P. T. Oak, and D. P. Barnard 4th.

[17] See *Proceedings of the American Society for Testing Materials*, vol. 32, pp. 772-774, 1932; Tentative Standard Viscosity-Temperature Chart for Liquid Petroleum Products, being A.S.T.M. designation D341-32T.

[18] See S.A.E. Journal, vol. 28, pp. 234-239, February, 1931; Cold-Starting Characteristics of Automobiles, by A. J. Blackwood and N. H. Rickles.

[19] See S.A.E. Journal, vol. 29, pp. 210-214, September, 1931; Results of Cold-Starting and Consumption Tests, by C. M. Larson, being discussion of the paper on Oil Consumption as Affected by Engine Characteristics, by H. C. Mougey.

[20] See S.A.E. Journal, vol. 29, pp. 141-147, August, 1931; Low-Temperature-Starting Development of Automobile Engines, by P. J. Kent.

[21] See S.A.E. Journal, vol. 33, p. 24, August, 1933; Oils.

[22] See S.A.E. Journal, vol. 33, p. 49, October, 1933; Lubricants.

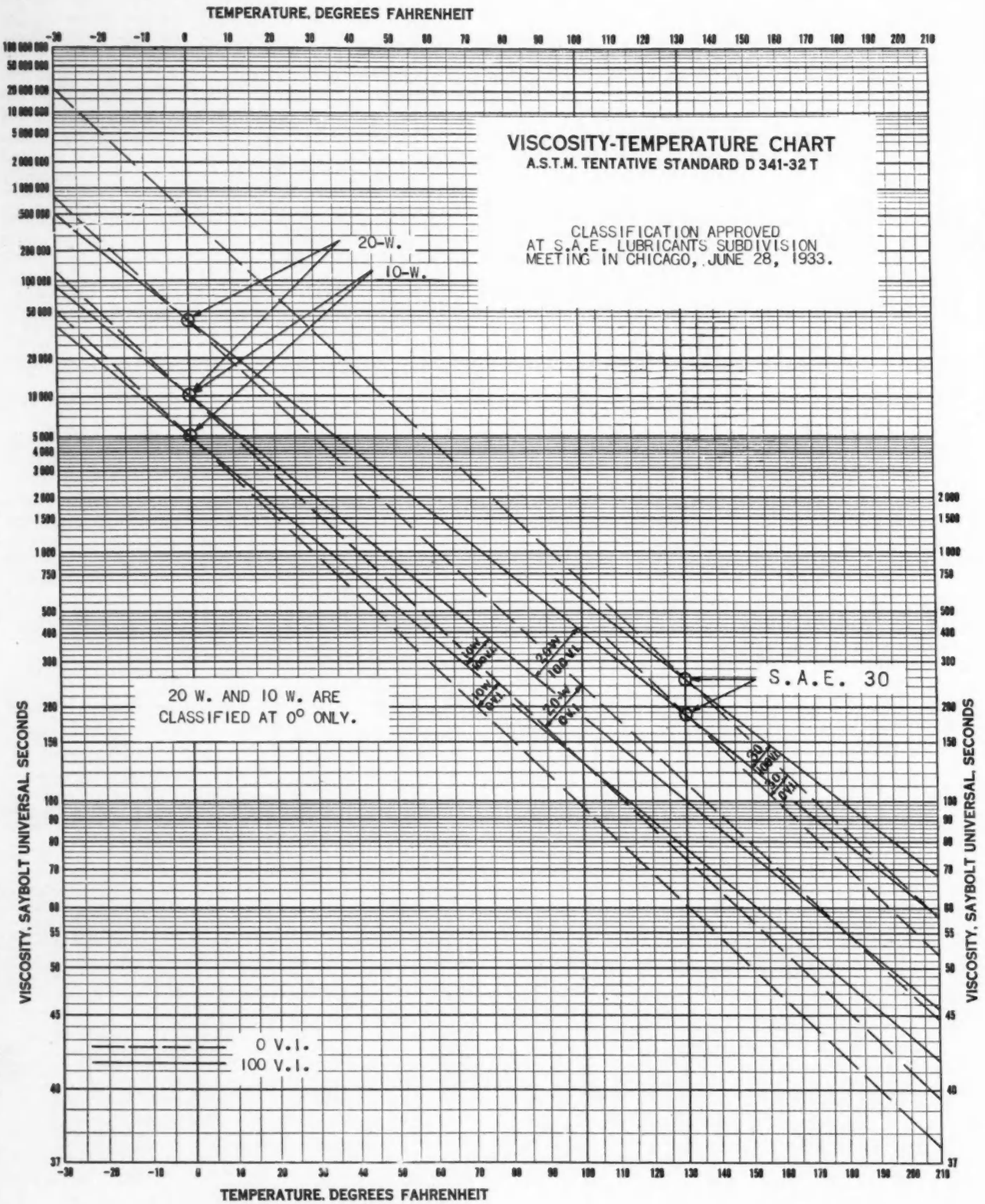


Fig. 2—Differences between 10-W and 20-W Oils and the Classifications S.A.E.-10, S.A.E.-20 and S.A.E.-30 Oils

It should be pointed out that some S.A.E.-20 oils come within the 20-W range at 0 deg. fahr. and can be classified as 20-W. In a similar manner some S.A.E.-10 oils may be 10-W, but unless the oils are classified and sold as 20-W or 10-W the individual car owner has difficulty in distinguishing between the oils that will permit starting and those that will not.

Sometimes the suggestion is made that, if automobile engineers would only put bigger and better starters and batteries on their engines, it would be possible to start on any oil the car owner might happen to have. However, as previously mentioned, J. H. Hunt pointed out, in 1924, that such large batteries and starters were not commercially possible, and the same statement holds true at the present time when engines are being built with a larger number of cylinders that have closer tolerances.

The chart in Fig. 3 shows how the torque required for cranking depends upon the friction resulting from the viscosity of the oil used. The engine used in this test was a six-cylinder 3 5/16 x 3¾-in. engine with cast-iron pistons. Using an S.A.E.-30 oil with a viscosity of 85,000 at 0 deg. fahr. (217 at 130 deg. fahr.) the torque required to crank 50 r.p.m. at 50 deg. fahr. is 28.1 lb-ft., and the torque available from the battery and starter is 80.1 lb-ft., almost three times as much as is required. However, when the temperature drops to 0 deg. fahr., the torque required to crank the engine with the S.A.E.-30 oil is 208 lb-ft. and the torque that can be supplied by the battery and starter is only 62.4 lb-ft. In order to crank 50 r.p.m. the starter and battery would have to be increased to deliver three and one-third times as much torque. By using a 10-W oil (8500 at 0 deg. fahr.) the torque required is only 44.7 lb-ft. and the starter and battery are seen to be able to deliver one and one-half times as much torque as needed. It is evident that increasing starter and battery equipment that can already deliver three times as much torque as is required in summer with a summer oil and one and one-half times as much as is required in winter with a winter oil, to enable it to deliver ten times its summer requirement so that a summer oil can be used in winter is not a commercial solution of the problem. If we assume some other cranking speed, for example, 25 r.p.m., the numerical values are changed but the story remains the same.

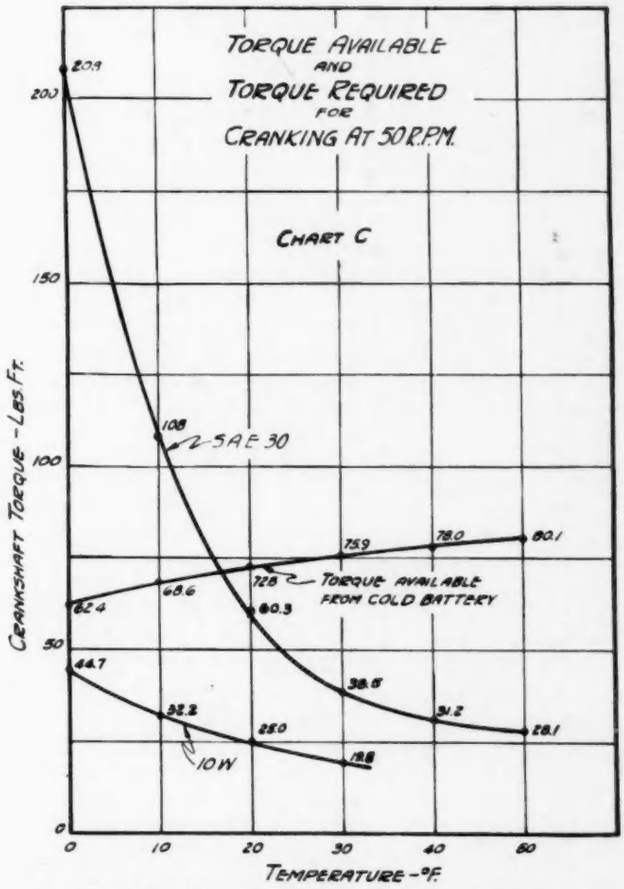


Fig. 3—Chart Showing How the Torque Required for Cranking Depends upon the Friction Resulting from the Oil Used

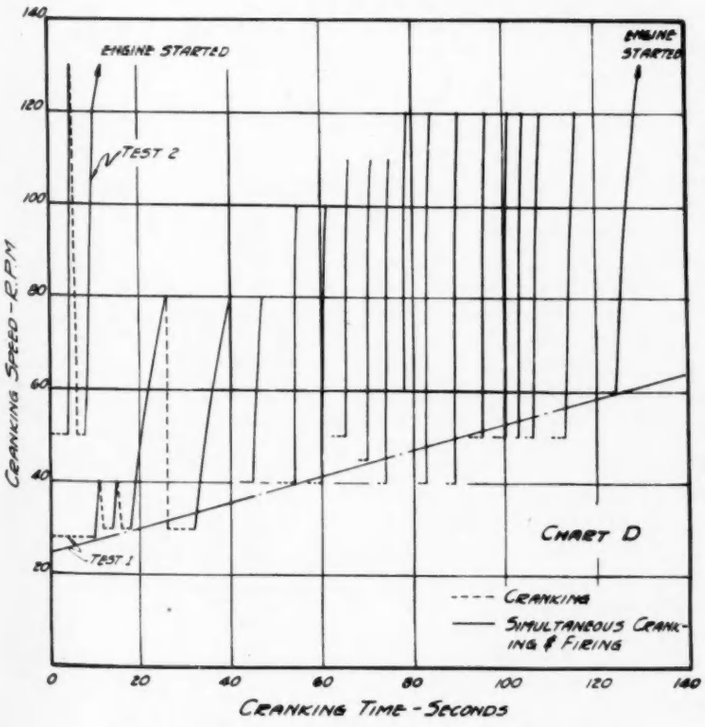


Fig. 4—Chart Resulting from Plotting Cranking Speed versus Cranking Time

Since it is a physical impossibility to get the public to change oils to meet the varying requirements of differences in hourly temperature or in driving speed, it is evident that an ideal solution to the problem would be for the cars to operate in winter, regardless of the weather or driving speed, on a single oil. As an illustration, the public has found that if an attempt is made to add anti-freeze to the radiator each day to prevent freezing at the temperature that will be reached in the night, incorrect guesses are likely to be made and freezing of the radiator will occur. By using in the radiator an anti-freeze sufficiently low in freezing point to afford protection on the coldest days that are expected, and which will also be satisfactory for high-speed driving on warm days in winter, this trouble can be avoided. In a similar manner, we believe that the oils used in automobiles should be such that the engines can be started on the coldest days likely to be experienced, and yet that they should operate satisfactorily at high speed on the warm days that always occur in winter.

The engine crankcase-oil must perform three major functions, which in the order of their importance are; (*a*) to furnish adequate lubrication, (*b*) to permit easy starting, and (*c*) to give reasonable oil mileage.

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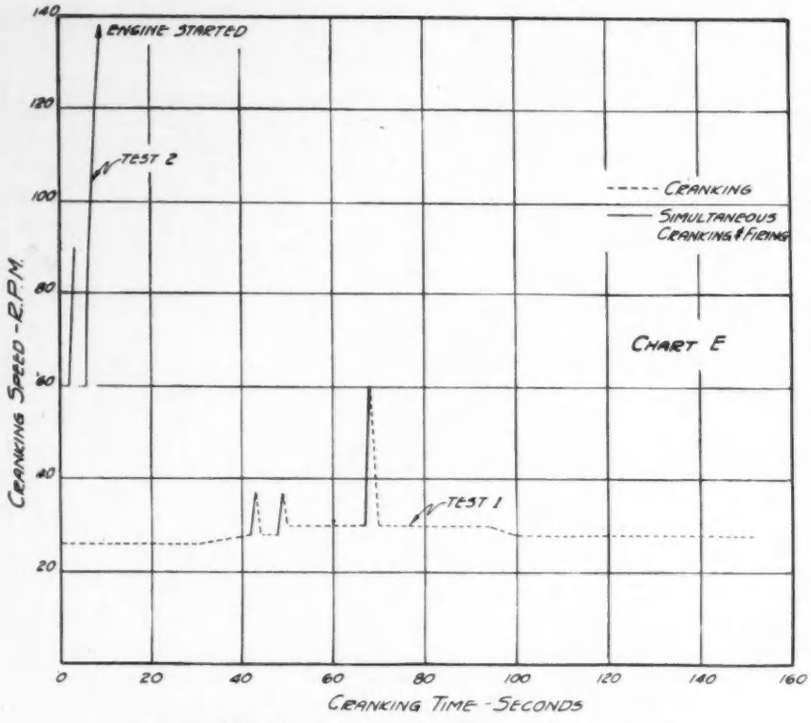


Fig. 5—A Similar Chart to That Shown in Fig. 4, But from an Engine of Somewhat Dissimilar Design

Since most people seem to pay most attention to oil mileage, it will be discussed first. Tests made by the Standard Oil Co. (Indiana) at the Indianapolis track[23] showed that, although higher-viscosity oils do enable the automobile to go more miles per gallon of oil at a certain speed, yet the difference is not as great as might be expected and the higher consumption of oil of lower viscosity may be offset in cost by the saving resulting from the decreased amount of gasoline required. The averages for a fleet of 13 makes of 1931 passenger cars for 1000-mile test-runs made at 55 m.p.h. are shown in Table 5.

Tests made at the General Motors Proving Ground, using several cars of the medium-price range made by different manufacturers, showed that, for a uniform sustained speed of 35 m.p.h., the average values in oil and gasoline miles per gallon were as shown in Table 6.

The average values used in Table 6 could be varied over a wide range and the conclusion would still be unchanged. That is, the gasoline cost is much greater than the oil cost, and a small saving in gasoline economy will offset a large difference in miles per gallon of oil. In addition, the total difference in cost of oil plus fuel is too small to be of importance to anyone except a fleet operator with a large number of cars. Even in the case of a fleet operator the great advantage in using lower-viscosity oils is in avoiding starting troubles rather than in the saving in cost of fuel plus oil which the lower-viscosity oils show.

When oil consumption is determined at different car speeds, it is found that, although the effect of oil viscosity is still apparent, by far the greatest single factor affecting oil consumption is speed. In 1922 Bull[24] called attention to the decrease in miles per gallon of oil at higher speeds; and, in 1928, Bramberry[25] presented curves showing this relationship. The tests[26] made by the Standard Oil Co. (Indiana) showed that as an average of 1000-mile tests of 13 makes of 1931 automobiles, the oil consumption at 55 m.p.h. was 6.9 times that of the same cars with the same oils at 30 m.p.h. This ratio varied, however, in the different cars, from as low as 2.3 to as high as 19.8.

Many tests made by various laboratories have shown the same effect of speed on oil consumption and there are numerous references in the literature and in the publications of the Society to this decrease in miles per gallon of oil at higher speeds. In studying some of these old records, and comparing them with those obtained more recently, it is encouraging to note that automotive engineers in successive models have been able to increase the miles per gallon of oil at a faster rate than the maximum speeds of the cars have been increased. In other words, although more oil is consumed at higher speeds, the miles per gallon of oil of cars of present design, at present speeds, is greater than it was several years ago when much lower speeds were the rule. As the various factors of designs of engines and lubricating systems, including pistons and piston rings, are better understood, it seems reasonable to predict that still further gains in miles per gallon of oil will be made, even using low-viscosity oils at very high speeds.

The second function which crankcase oil should perform—permit easy starting—is one which has been recognized since starters were first invented. The cranking speed versus cranking time for two engines of somewhat dissimilar design is shown in the charts in Figs. 4 and 5. In Fig. 4, Test 1, the initial cranking speed during the first 10 sec. was 28 r.p.m. The first fire occurred at 10 sec., but the cranking speed, due to the use of high-velocity crankcase-oil, was too low to distribute a combustible mixture to all cylinders and while the cranking speed increased to 40 r.p.m. the engine failed to continue to run. Cranking was continued for approximately 2 min. during which time, due to dilution from unburned fuel and the heat generated through friction, the cranking speed increased to 60 r.p.m. At this cranking speed the engine was able to continue to run after the first fire. In Fig. 4, Test 2, using a lower-viscosity crankcase oil, and with an initial cranking-speed of 50 r.p.m., the first fire was obtained in 4 sec. and the engine continued to run after 8 sec.

In Fig. 5, Test 1, with an initial cranking-speed of 26 r.p.m., practically no fuel was introduced onto the cylinder walls with the result that the cranking speed remained almost constant during 2½-min. cranking. With a lower-viscosity crankcase-oil as in Test 2 and with an initial cranking-speed of 60 r.p.m., the first fire occurred in 2 sec. and after 6 sec. of cranking the engine continued to run. These tests show very conclusively the necessity for using a low-viscosity crankcase-oil which will allow a high enough initial cranking-speed to permit starting within the first few seconds so as to prevent

[23] See S.A.E. JOURNAL, vol. 29, pp. 208-209, September, 1931: Should Acquaint Car Owners with the Facts, by R. E. Wilkin, being discussion of the paper on Oil Consumption as Affected by Engine Characteristics, by H. C. Mougey.

[24] See S.A.E. JOURNAL, vol. 10, pp. 513-522, June, 1922; Oil Consumption, by A. A. Bull.

[25] See S.A.E. JOURNAL, vol. 23, pp. 495-498, November, 1928; Pistons and Oil-Trapping Rings for Maintaining an Oil Seal, by Harry M. Bramberry.

[26] See S.A.E. JOURNAL, vol. 29, pp. 208-209, September, 1931; Should Acquaint Car Owners with the Facts, by R. E. Wilkin, being discussion of the paper on Oil Consumption as Affected by Engine Characteristics, by H. C. Mougey.

an excessive drain on the battery.

These data indicate how light oils give easy starting and they are checked by experience in service. At a meeting of Packard Service Managers from all large cities held in March, 1934, cold-weather starting-difficulties were discussed and the managers from New York, Chicago, Philadelphia and Boston were greatly elated at the reduction in cold-weather starting-complaints. Several made the statements that starting complaints have easily been reduced one-half and that 10-W and 20-W oils were absolutely essential for them from now on. The question was asked of these men if they had received or knew of any complaints from the use of these new winter-oils and none were made, an unusual condition to exist with service managers. This statement of fact, while not a numerical value, is of extreme importance and indicates that 10-W and 20-W actually meet service conditions.

The third and most important function the crankcase oil must perform—furnish adequate lubrication—is probably the one about which there is the most mystery in the minds of the public. The practice of sacrificing easy starting, fuel economy, maximum speed, and other desirable properties to obtain more miles per gallon of oil, has made many people actually afraid to use oil of the proper viscosity for winter.

Tests made by the Perfect Circle Piston Ring Co.[27], using a series of test oils supplied by the Standard Oil Co. (Indiana), showed that "blow-by" with all the oils tested was greater at 60 m.p.h. than it was at 50 m.p.h. with any of the test oils. However, at neither of these speeds was "blow-by" objectionable in amount, thus indicating satisfactory sealing. It is interesting to note that these oils varied from 39.5 at 210 deg. fahr to 72 at 210 deg. fahr. and included oils from different crudes varying in viscosity index and volatility.

There are many records in the automotive publications showing that oils which are low in viscosity are still able to furnish adequate lubrication. The report of tests on cylinder wear made by the British Institution of Automobile Engineers[28] is very interesting, even though the test conditions were only 1600 r.p.m with a brake mean effective pressure of 59 lb. per sq. in. They found, however, that under these conditions with clean oil they could add kerosene to the oil used (Mobiloil BB) until they had 90 per cent dilution before the rate of cylinder wear increased. By operating under very dirty and dusty conditions, more wear was obtained, even with fresh full-bodied oil, but the rate of wear was not increased by dilution until 75 per cent dilution was exceeded. A 90-per cent kerosene-dilution with Mobiloil BB gives a viscosity of 36 at 100 deg. fahr. and 75-per cent dilution gives a viscosity of 40 to 100 deg. fahr. The oil-sump temperatures for their test conditions at a speed of 1600 r.p.m. are not recorded, but they are probably in the neighborhood of about 140 deg. fahr. The values 36 and 40 at 100 deg. fahr. are approximately 33 and 36 at 140 deg. fahr., respectively.

While laboratory tests are of interest, the real answer, of course, is performance in service. One of the most spectacular service records, showing the ability of 20-W oil to properly lubricate under severe operating conditions, occurred when Ab Jenkins made his record-breaking performance of driving a modified stock Pierce-Arrow car under summer desert-heat conditions 3000 miles in 25½ hr., averaging 117.98 m.p.h., the engine speed at times being as high as 4590 r.p.m. This run was made at Salduro, Utah, in August, 1933. The engine was lubricated with Pennzoil S.A.E. 20, a winter oil, within the limits of 20-W, although the cans in which the oil was sold were marked S.A.E.-20. As previously pointed out, some S.A.E.-20 oils also come within the classification limits of 20-W, and such oils may be sold as either S.A.E.-20 or 20-W. This oil was used because previous tests had shown that it lubricated the car satisfactorily and, on account of the fact that it offered less friction to the moving parts as compared with the next higher viscosity, S.A.E.-30, it permitted a higher maximum speed. Since the weather in the winter may vary from very low temperatures to warm days in which summer temperatures are approached, it is apparent that the oil which is used in an automobile in winter should permit starting at the lowest temperature likely to be encountered and yet it should provide safe lubrication under high-speed high-temperature conditions on the warmer days. The use of the oil meeting the limits of 20-W by the Pierce-Arrow car in the Salt-Lake test certainly indicated that ordinary drivers are not taking chances in using 20-W for the most severe road service likely to be experienced in winter.

This performance has received much publicity and it should be convincing evidence to anyone that 20-W oil will properly lubricate a car under any conditions, but many more examples could be given to substantiate the statement that 20-W or 10-W oils will properly lubricate an engine under the conditions of service for which their use is recommended. All new Packard cars leaving the factory, both summer and winter for the last ten years, have had an oil in the crankcase meeting the limits of 20-W. All Packard-12 cars are driven at the Packard Proving Ground track for 250 miles at a speed from 50 m.p.h. to maximum before delivery, and this running is all done on oil meeting the classification 20-W. This track is illustrated in Figs. 6, 7 and 8. Two Packard engines have just completed 10,000 miles each on the dynamometer, running at 60 and 70 m.p.h. with road-load torque using 20-W oil, controlled at 200 deg. fahr., without appreciable wear of cylinders, pistons, rings or bearings. One of these engines was then run 2000 miles with 10-W oil under the same test conditions and is now back on the 20-W oil-test without showing any troubles. For the last ten years from 50 to 75 Packard employees have kept their own private cars in a garage where all fuels and lubricants have been furnished by the factory and where all repairs have been made. With the exception of the last four years an oil meeting the 20-W classification was used both summer and winter with perfect satisfaction. S.A.E.-30 lubrication was adopted four years ago for summer use to reduce oil consumption;

Table 5—Oil Mileage for 1931 Passenger Cars

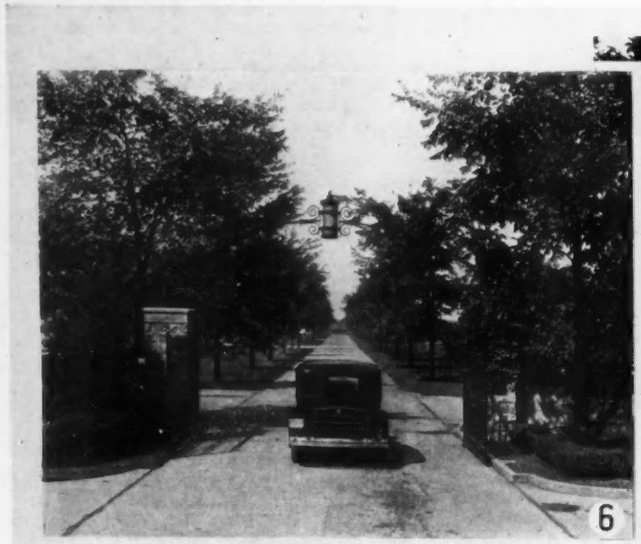
S.A.E. Oil No.	Viscosity, Sec. at 210 Deg. Fahr.	Miles per Gallon of Oil	Gasoline Consumption, Per Cent
30	58	406	93.6
50	83	562	...
60	107	707	100.0

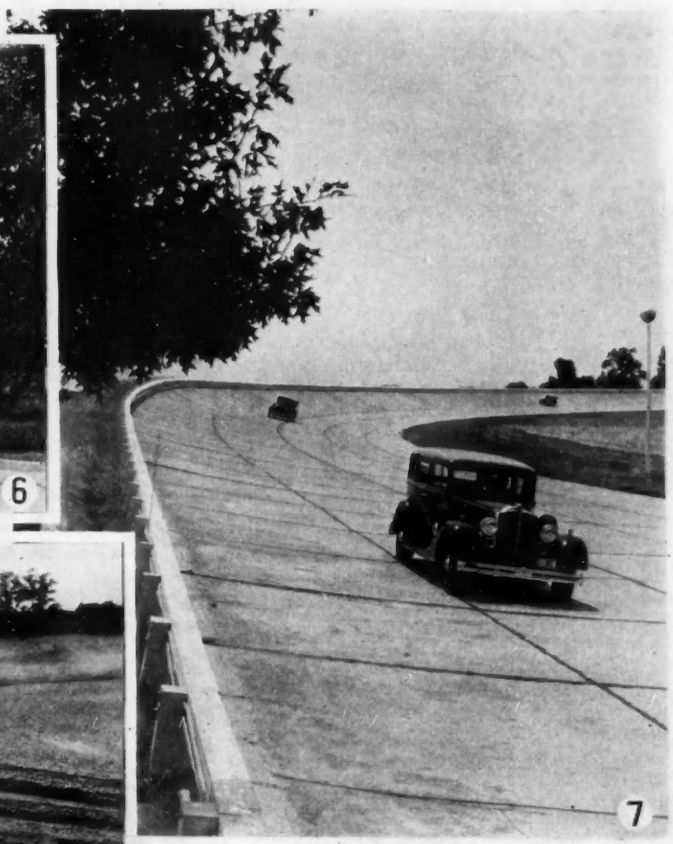
Table 6—Oil Mileage for Medium-Price Cars

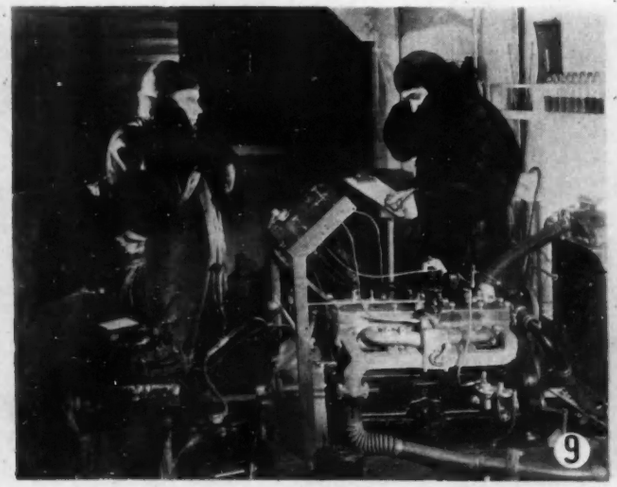
Items	Oil 10-W	Oil S.A.E.-30
Miles per Gallon of Oil	1,500	2,500
Miles per Gallon of Gasoline	13.4	13.0
Gallons of Oil per 10,000 Miles	6.67	4.0
Gallons of Gasoline per 10,000 Miles	746	769
Cost of Oil at $1.00 per Gal.	$6 67	$4.00
Cost of Gasoline at $0.20 per Gal.	$149.20	$153.80
Extra Cost for Oil	$2.67	.....
Saving in Cost for Gasoline	$4.60	.....
Net Saving per 10,000 Miles	$1.93	.....

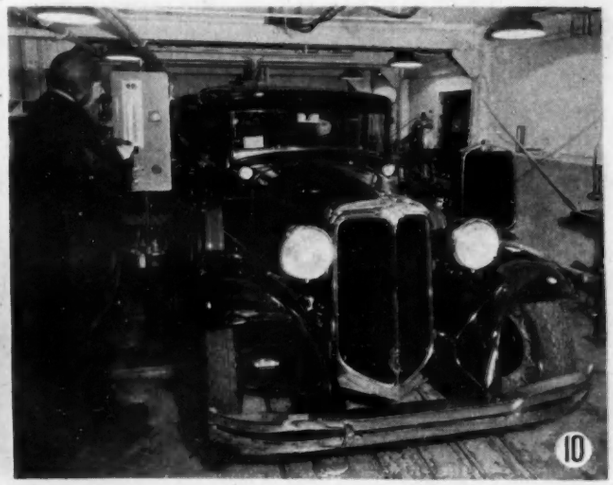
[27] Data from a private communication to the authors.

[28] See *The Engineer*, (London), vol. 155, pp. 634-636 and pp. 660-662, June 23 and June 30, 1933; Cylinder Wear, being a Report by the Institution of Automobile Engineers.









Figs. 6 to 10—Fig. 6—Entrance to the Packard Proving Ground at Utica, Mich. Fig. 7—A Packard-12 Car on an Initial-Test Run at Top Speed on the Packard Proving Ground, Using 20-W Oil. Fig. 8—Severe Testing of a Car on the Packard Proving Ground, Using 20-W Oil. Fig. 9—An Engine Test on the Chrysler Corp. "Cold-Room" Dynamometer. Fig. 10—A Chassis Dynamometer-Test in the Chrysler Corp. "Cold Room"

20-W oil is still used during the fall, winter and spring months.

Fleet-operation records are of value in that they represent an average of a large number of operating units or cars. During the winter of 1932-1933, the fleet of the Dayton Power and Light Co., operating 200 pieces of equipment in Southern Ohio, used an S.A.E.-20 oil in all their equipment. The records show that, during the occasional short periods of zero weather experienced in that winter, considerable difficulty was experienced in starting those cars that were not kept in the heated garage. Ordinarily, this fleet changes from winter to summer oil in May, and in May, 1933, S.A.E.-30 oil replaced the S.A.E.-20 oil used during the winter. From May 1 to 15, the fuel economy of the fleet, instead of improving as would have been expected with the warmer weather, showed a pronounced slump, and it was decided to discontinue the use of S.A.E.-30 and return to S.A.E.-20 in an attempt to improve the gasoline economy. For the last half of May, when S.A.E.-20 was in use, the gasoline economy of the fleet picked up to a marked extent, and it was decided to continue throughout the summer months with the use of S.A.E.-20.

In October, this fleet had in the past changed from summer to winter oil. At this time, the transportation supervisor consulted the Chrysler Engineering Division, and, on its recommendation, went to a 10-W oil for use during the winter months. The 10-W oil was put in in December, and although the temperature for that month averaged 4 deg. lower than the temperature of November, 1933, the gasoline economy showed an appreciable improvement. The use of 10-W oil was continued throughout the winter months and, for the first time in the history of this operation, no starting difficulty was experienced with the cars kept in unheated garages. The only two cases of starting difficulty on record for the winter of 1933-34 developed in two cars that were using S.A.E.-20 oil and had not come in for their recommended change to 10-W. Incidentally, the weather at times during this period was abnormally cold and a 20-year low-level was reached in February, and still no starting difficulty was experienced. In going to 10-W oil, the main consideration was that of increased gasoline economy, and it was anticipated that the fleet would experience somewhat greater oil usage. However, a check of the months of December, January, February and March, indicates that the oil usage was not appreciably greater than that of past years when S.A.E.-20 oil was used. At most there was not greater than 5-per cent variation from previous years. It is the intention of this fleet to use during the summer months an S.A.E.-20 (or 20-W oil) and change to 10-W for the winter operation extending from November to May inclusive.

Even though 20-W is usually recommended for most of the winter-temperature conditions encountered in the Detroit area, the low viscosity of 10-W oil is needed during the occasional sub-zero spells for satisfactory starting. The 10-W oil is satisfactory during winter as is evidenced by the experience of seven employees of the Chrysler Engineering Department who used 10-W in their personal cars this last winter. Most of the driving was done at moderate speeds, but the records show considerable driving done at 50, 60 and 70 m.p.h. The seven cars represent a total mileage of 18,595 miles, with an average oil consumption for the period of 3686 miles per gal. Needless to say, no starting troubles were experienced. Engine and chassis tests on the dynamometers in the "cold room" of the Chrysler Corp. are illustrated in Figs. 9 and 10.

At the General Motors Proving Ground tests were run in August, 1933, on eight automobiles made by different manufacturers. Oil-consumption tests were made using five different oils. During this series of tests over 1200 miles were driven at a sustained speed of 65 m.p.h. for 100-mile test-periods with a low-viscosity-index 10-W oil giving the lowest viscosity at the high operating temperatures which would be experienced with any of the commercial 10-W oils. All of these cars of eight makes operated satisfactorily on this 10-W oil.

A 1933 eight-cylinder General-Motors car was then put on the road using 10-W oil with instructions to the driver to operate at maximum speed as much of the time as possible. At the end of 1500 miles the engine was still satisfactory, but the test was then changed to a 1934 car of the same make as it was felt the information would be of more value. The test was discontinued at the end of 6000 miles without any difficulty having been experienced with the 10-W oil. Two eight-cylinder General-Motors cars were then put on the track on a 14,500-mile test on a schedule intended to represent the most severe owner-driver service. These cars were driven 600 miles per day including 90 miles per day at 80 m.p.h. made in three 30-mile test-periods. At the end of this test the bearings, cylinders, pistons and rings were in excellent condition. In addition, many other test cars have been driven under all types of service conditions with 10-W oils without experiencing difficulty. As previously mentioned, the use of lower-viscosity oils means a greater consumption of oil regardless of the operating conditions. At high speeds these low-viscosity oils may give lower miles per gallon of oil than the owner might anticipate. In using these low-viscosity oils, the oil level in the crankcase should be checked frequently and the level should be maintained.

Summary on Winter Lubricants

(1) The information in regard to the various factors involved in winter starting and operation has been available for many years, but it has not been generally known or appreciated. This lack of general information interfered with the classification, recommendation and use of suitable winter oils.

(2) At present, the various factors are well enough known by the oil and automotive industries to permit the classification, recommendation and use of suitable winter oils.

(3) To permit cold starting, the engine must be cranked at a high enough speed to permit satisfactory distribution of the gasoline-air mixture to the cylinders.

(4) Oils 10-W and 20-W, classified in accord with their viscosity at 0 deg. fahr., are required to keep the cranking friction low enough to be within commercial limits.

(5) Oils 10-W and 20-W, which are necessary for cold-weather starting, are satisfactory for use throughout the entire period of time for which they are recommended, including the high-speed driving on the warm days that are likely to be experienced in winter.

(6) The lower-viscosity oils do not give as many miles per gallon of oil as do the higher-viscosity oils, but the difference in oil consumption, due to viscosity, is not as great as many people believe.

(7) The greatest factor affecting oil consumption is engine speed.

(8) The lower-viscosity oils, on account of causing less friction, will give more miles per gallon of gasoline, and the total cost of oil plus gasoline is usually less with low-viscosity oils than with high-viscosity oils.

Dynamics of the Modern Automobile

By George L. McCain
Research Engineer, Chrysler Corporation

MR. McCAIN enumerates in the simplest possible form some outstanding features of the new automobile-dynamics, and gives practical commercial reasons for his conclusions.

As referred to in the paper, streamlining is the reshaping of bodies to reduce air resistance at a commercial speed of about 45 m.p.h. An analysis of this subject is presented, and the effects of a redistribution of passengers and units are discussed. Riding-quality model-test results and weight distribution are commented upon. A bibliography of streamlining is included.

The overdrive is considered by Mr. McCain as part of the airflow car, and the curves show that —with the overdrive—a greater car speed can be reached under favorable wind-conditions, exceeding the curve values, without an excessive engine-speed. Forces acting upon planetary overdrive gears are treated in the Appendix.

Much remains to be done toward reduction of wind resistance, in Mr. McCain's opinion. He says that an ideally streamlined body would be too long to park and turn sharply in city traffic, and would involve an extra amount of weight that would constitute a drag against acceleration. Further, that a long tail involves more waste space per wheelbase, and, with the engine in front, does not adapt itself to readily usable baggage compartments.

THE modern automobile, someone has said, is individual transportation to go from here to there, and on indefinitely, as fast and as comfortably as possible. Over the last few years, road building has advanced and traffic control has developed to a stage such that greater speed can be safely used and sustained. High-speed travel with the cars of yesterday, in spite of salesmen's promises of an optimistic economy, is expensive in fuel and oil, and modern conditions call for a new combination.

The World War, with its many aviation activities, and aviators who have since turned their skill toward commercial aviation, have created in the public's mind a responsiveness to streamlining to lessen resistance to travel through the air. Decrease of wind noise is a real welcome development and contributes greatly to comfortable travel at high speed. Another element of annoyance is high engine-speed and, even aside from a desire to extend engine life, the experienced driver will avoid sustained high engine-speed. Two-speed axles and twin high-gears have been used in an attempt to alleviate this condition, but the average driver objects to shifting or does not know when and where to use the extra gear. Gears that are not almost silent are not used through long periods of running, even though the advantages and economy of slower engine-speed are well known.

High-speed travel over the average road magnifies the irregularities, which are passed over almost unnoticed at 30 to 40 m.p.h. The problem of suspension, therefore, is of equal, if not greater, importance than that of air resistance or streamlining. This involves springs, shock absorbers, and passenger position. The rear-seat passenger, who should have at least as comfortable a position as the one in front, has had a number of things happen to him. Tires have become bigger to improve riding qualities, and the tread increased some to avoid pinching the rear seat to a two-passenger width. Even so, some room is lost; the wheel housings have become more or less parts of the arm rests, and have made weird shapes of rear cushions. Low cars are popular because they are safer, and this condition has forced another bump into the rear cushion, above the rear-axle housing.

A number of conditions have brought about, therefore, a new deal for the rear passenger. In the development we will discuss, we will start with the rear passengers where they should be and build up a structure around them, trying at the same time to incorporate a design which will furnish com-

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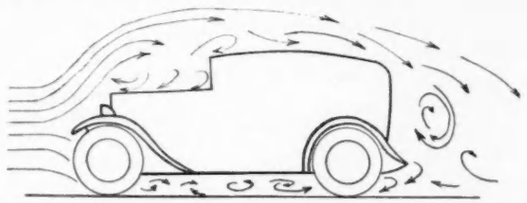

Fig. 1—Graphical Representation of the Flow of Air in the Vertical Plane, for the Conventional Car

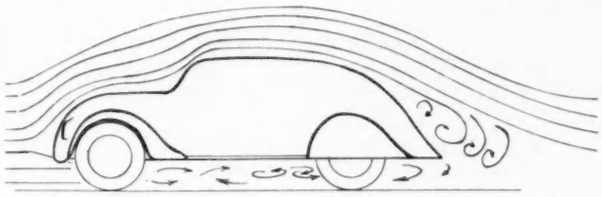

Fig. 2—Graphical Representation of the Flow of Air in the Vertical Plane, for the Chrysler Airflow Car

fort, stability, and endurance for the machine as well as for the passenger. We must do this commercially for the producer and the consumer and not penalize the maintenance problem.

Streamlining

First, let us deal generally with the question of streamlining and define it for our general understanding. As referred to in this paper, streamlining is the re-shaping of bodies to reduce air resistance at speeds which are deemed commercial; that is, approximately 45 m.p.h.

In speaking of streamlining we have borrowed from the aeronautical industry a widely used term, "air resistance," designated as R_a, which is the air resistance to motion offered by the body. This resistance to motion varies as the density of the medium termed ρ, the area of greatest section of the body, termed A, and as the square of the velocity, designated as V, as expressed by the formula $R_a \propto AV^2$ (proportional).

As a proportionality factor is required, we may write $R_a = KAV^2$. This may be expressed as pounds of drag equals a coefficient K, times the maximum cross-sectional area of the body, taken in a plane perpendicular to the direction of motion, expressed in square feet, times the velocity in miles per hour squared. Here, K is a quantity including the air density, and expresses pounds of air resistance per square foot of maximum cross-sectional area per mile per hour at standard atmospheric conditions taken as 59 deg. fahr. and 29.921 in. of mercury. We may also write $K = R_a/AV^2$, as K is plotted as the abscissa for some of the following curves. Anyone interested in a more complete discussion of this fundamental formula is referred to any standard book on aerodynamics.

Of the total car resistance, rolling resistance and air resistance, air resistance rapidly becomes of increasing importance at speeds above 35 m.p.h. The air resistance may be classified as skin friction and eddy-making resistance, of which eddy-making resistance, or form resistance as it is sometimes referred to, was of chief interest in our work. The now familiar comparison of the flat plate and the streamlined form is omitted, and we will consider the case of airflow around the body and the formation of eddy currents behind it.

The eddy formation in the rear was the first problem attacked. These eddy currents or turbulent-flow conditions occurring directly in the rear and around the body of the car have a relation between the shape of the body and the relative air velocity such that the more we depart from a true streamline form, that is, a solid body which produces approximately streamline flow, the greater is the energy imparted to the airstream to overcome the effect of these eddy currents. This energy is imported to the airstream by the moving body and is dissipated as unavailable energy, which means that to reduce air resistance we must reduce turbulent airflow, for such a reduction means a reduction in air resistance or R_a.

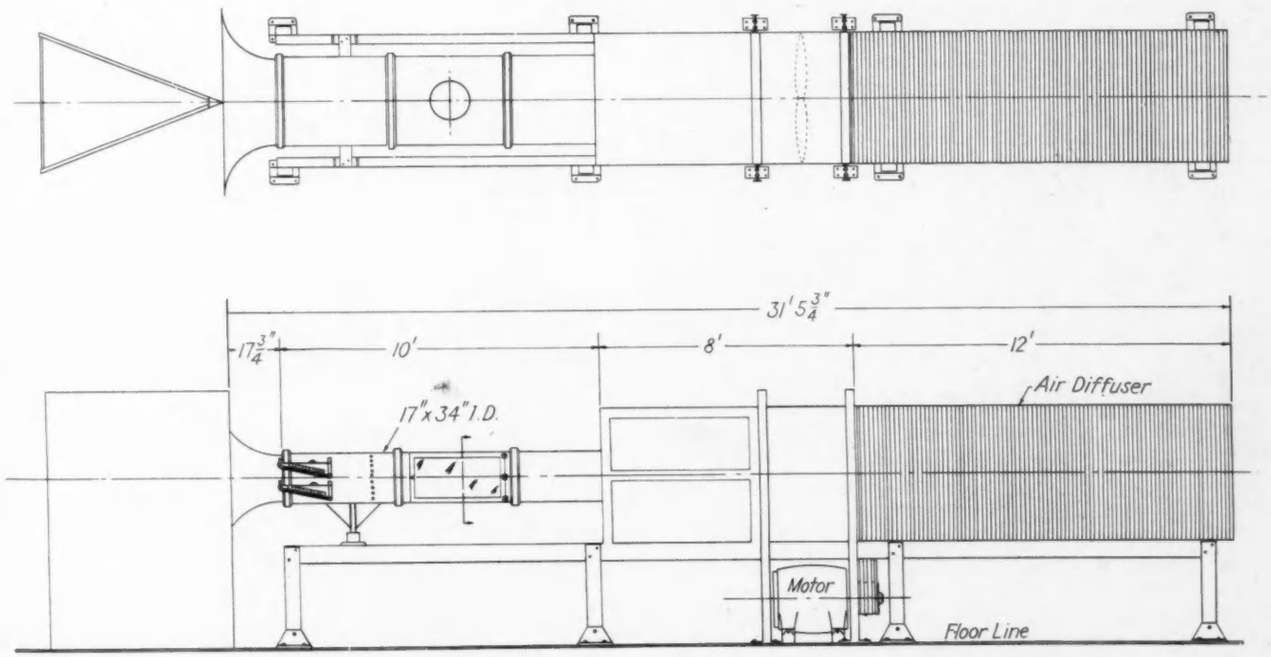

Fig. 3—Wind Tunnel in Which the Investigation of Airflow Characteristics Was Made

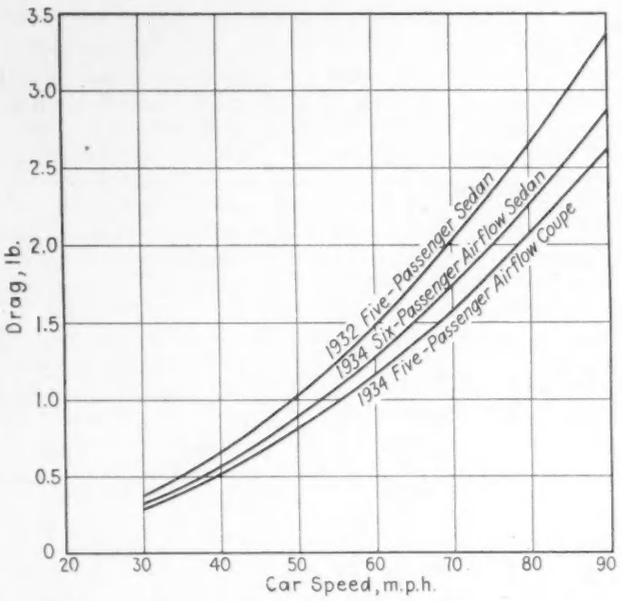

Fig. 4— Comparison of the Conventional-Type Car with the Airflow-Type Car

Theoretically, this means streamlining in both a horizontal and a vertical plane.

No particular attempt was made to reduce R_a in a horizontal plane, other than the slight bulging out of the body, filling in the spaces between the hood and the fenders, and mounting the headlights flush. In the vertical plane, a reduction in the turbulent air-space existing at the rear of the conventional car, such as is shown in Fig. 1, was attempted. Rounding the nose and sloping the rear deck in a vertical plane gives an airflow similar to that shown in Fig. 2. This type of rear deck is ideal for the moving of passengers and the engine forward, and is very advantageous to the riding qualities as will be shown later.

All the investigation pertinent to the airflow characteristics of this type of car was done in the company's own wind tunnel. Models to be tested were mounted on a flat plate and, while some might question this method of testing, the results were comparative. Fig. 3 shows roughly the arrangement used. Fig. 4 shows graphically a comparison of this design with the conventional car, and the following figures show some relative values obtained in wind-tunnel tests.

Model	Drag, lb. at 60 M.P.H.	Frontal Area, Sq. Ft.	K
1932 Sedan	1.50	0.255	0.001630
Airflow Sedan	1.28	0.258	0.001375
Airflow Coupe	1.17	0.253	0.001280

As previously stated, the desired sloping rear deck involved moving the passengers and chassis units forward. Let us start by building up an ideal passenger carrier, as, after all, that is what the modern motor-car should be. Fig. 5 shows the evolution very simply. Two seats are set up and wheels placed under them in the most advantageous place from a standpoint of riding comfort. Enough head room and a desirable luggage compartment determine the line of the top and rear deck. The location of the engine with respect to the passenger compartment gave the unit logical hood outline. In this arrangement, the engine had to be raised and the oil pan modified to clear the front axle, and the fan placed on the crankshaft because of the low radiator-position. A number of problems presented themselves. The radiator top tank had to be made in two parts, one of which is to one side and back of the radiator in the body-front extension.

Accessibility was a problem, but two developments helped very materially. Valves and valve seats have been developed to a state where very little attention is needed to keep them running properly. Furthermore, manifold design, heaters and the like, for years have been gradually blanketing over the valve covers, and the car of 1932 and 1933 had larger front fenders, altogether making any work on the side of the engine rather difficult. In the airflow car the movement of the engine up and forward places it squarely in line with easily removed plates, and, thanks to easily detachable wheels, the service job, though rare, is more easily done than before.

Fig. 6 shows the location of the center of gravity of the car units of a five-passenger 1933-model sedan. Fig. 7 shows how the units are arranged on the airflow model. The most

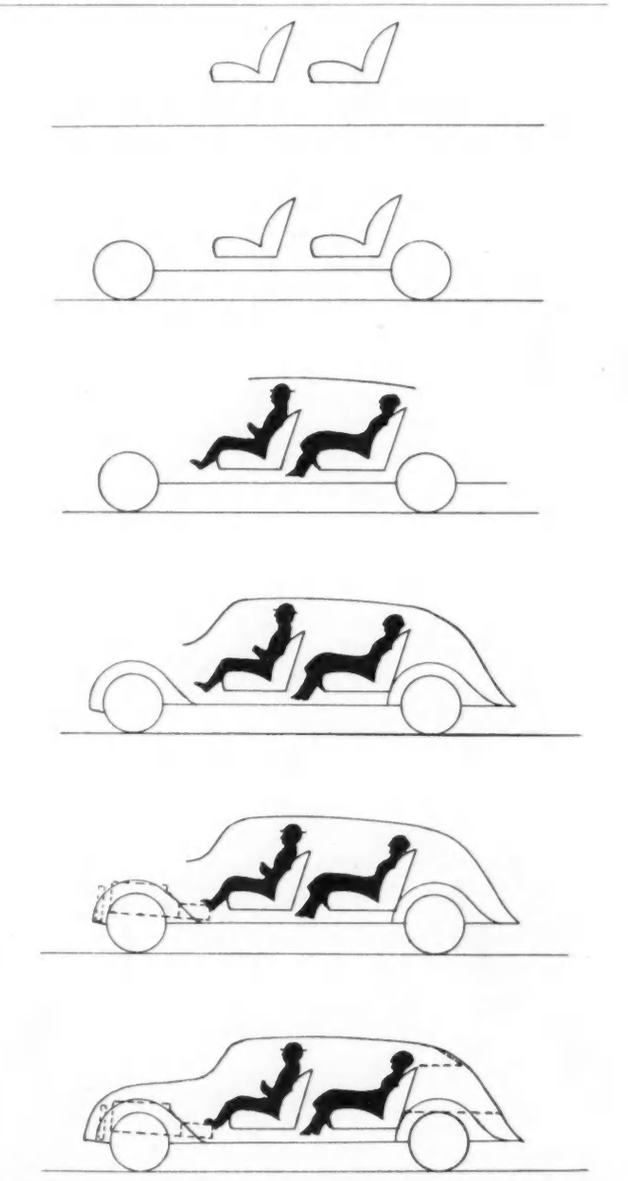

Fig. 5—Evolution of the Modern Motor-Car

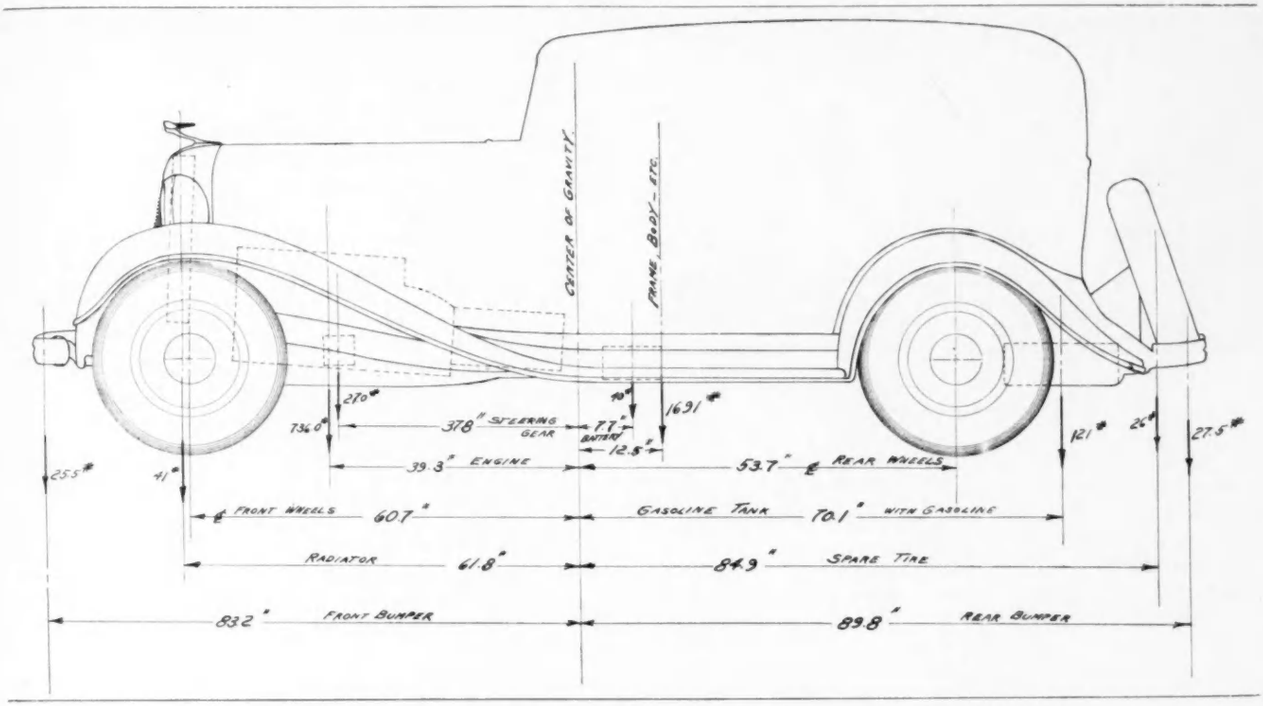

Fig. 6—Location of the Center of Gravity of the Units of a Five-Passenger 1933-Model Sedan

important factors are the engine and the body, as they have the greatest influence over the new weight distribution. The center of gravity of the engine was moved forward approximately 18 in. from the old position. The body itself was spread out more lengthwise, and that, together with the moving of the engine, made it possible to place the rear passenger seats in front of the rear axle and, at the same time, lower. This alone resulted in a better rear-seat ride, as the vertical movement of the passengers is smaller for any given angular movements about the front axle. But that is only one item entering into the consideration of the improvement of the ride of the airflow car. There are more important factors to be

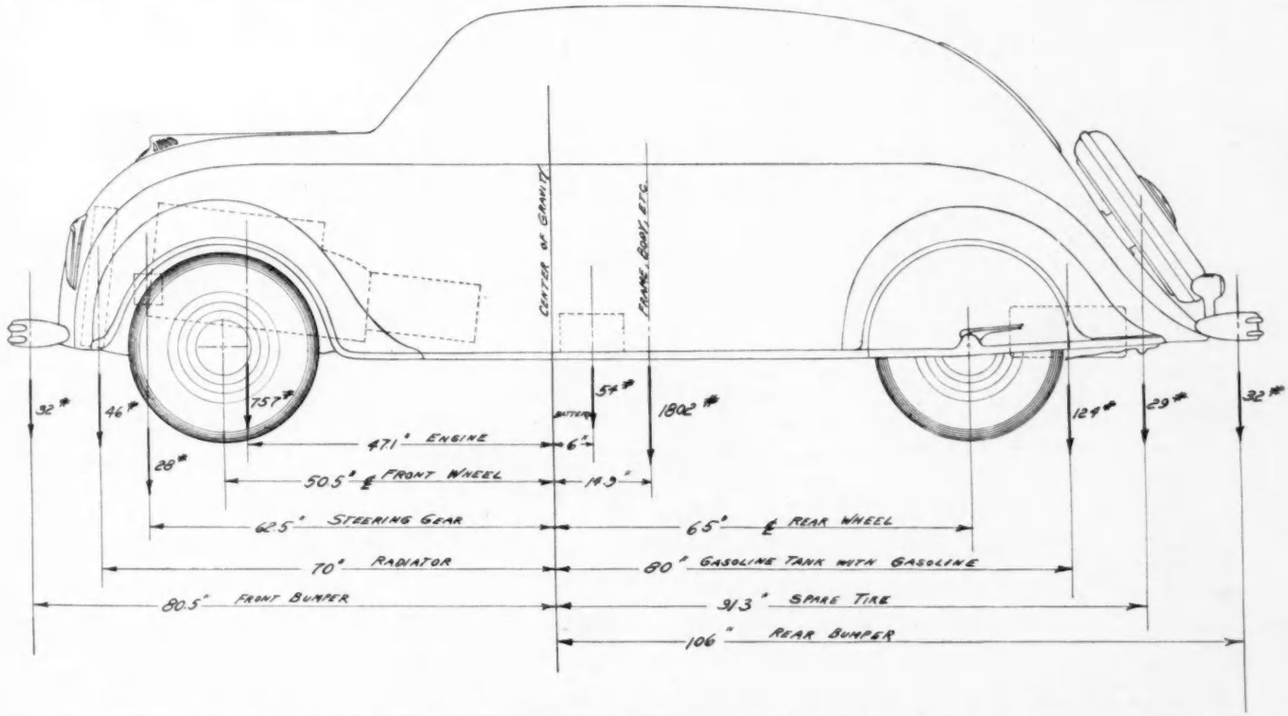

Fig. 7—Location of the Center of Gravity of the Units of an Airflow-Model Car

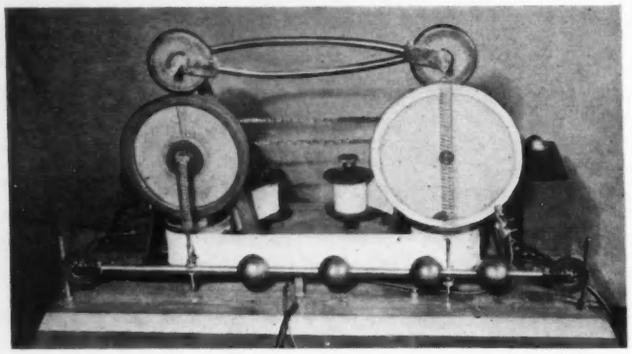

Fig. 8—Apparatus for Recording the Vibration of a Model That Is the Equivalent of a Car

considered, one of which is the concentration of more weight over the front axle. The front-spring rate was lowered somewhat, but the net result was a greater static deflection without sacrificing front-end stability.

The static deflection is the determining factor of the period of oscillation or frequency of the springs and the mass they are supporting. It can easily be shown how a weight suspended by a spring will have a high period when the spring is stiff and the weight small, and that it will have a lower frequency when the spring is soft and the weight is greater. The higher the frequency, the higher the acceleration imparted to the body will be, as the acceleration varies with the square of the frequency. This means that the acceleration will be higher; and the static deflection of the springs, lower.

It has been found that the riding comfort of passengers depends largely upon the accelerations imparted to them. If it were possible in the airflow design to increase the static deflection of the front springs more than had been practical heretofore with the rigid axle, the ride would be improved due to the lower frequencies resulting.

Taking the conventional sedan as an example again, we find that the static deflection of the front springs on that model was 3.19 in., which corresponds to a frequency of approximtely 115 cycles per min. for that car. By increasing the static deflection on the airflow car to 4.35 in., the frequency would have dropped to about 97 cycles per min. if the spring rate had only been lowered to give the new static deflection. The result would have been a frequency within the comfortable range of 70 to 100 cycles per min., but the front end would have been unstable.

The other factor governing the acceleration imparted to a body is its moment of inertia. The greater the moment of inertia, the lower the acceleration will be. By moving the weights out farther from the center of the car, that is, increasing the weight over the front axle and keeping the spring rate approximately the same, the period of oscillation was brought down to about 90 cycles per min. without sacrificing the least bit of front-end stability. The vertical accelerations are reduced some more, not only due to the fact that the frequency was lowered but also because the amplitude of movement has become considerably less.

Riding-Quality Model-Tests

Previous to the building of the actual first airflow test-cars, an investigation was conducted to determine the effect of the weight distribution upon the quality of the ride. For that purpose an apparatus was constructed as shown in Fig. 8, which recorded the vibration of a model. This model is the equivalent of a car and the weight distribution on it could be changed to give any desired values of i^2/AB. It was also so arranged that the vibrations caused by a bump on the front wheels only, or the rear wheels only, or both, could be recorded. It was fully demonstrated on that apparatus how a condition would be set up where no reaction on the rear spring due to front-spring excitation, or vice versa, occurred. This was the case when $i^2/AB = 1$.

Fig. 9 shows two typical charts obtained by letting the front wheel only hit the bumps. One is for $i^2/AB = 0.81$, corresponding to the weight distribution of a car of conventional design, and the other one for $i^2/AB = 1$, corresponding to the distribution of weights in the airflow cars.

Weight Distribution

The distribution of weights in the car has the most important effect on the riding quality because it governs the reactions on the rear springs when the front wheels go over an obstruction, or vice versa. When the actions of the front and rear springs are entirely independent of each other, a maximum in riding comfort is obtained.

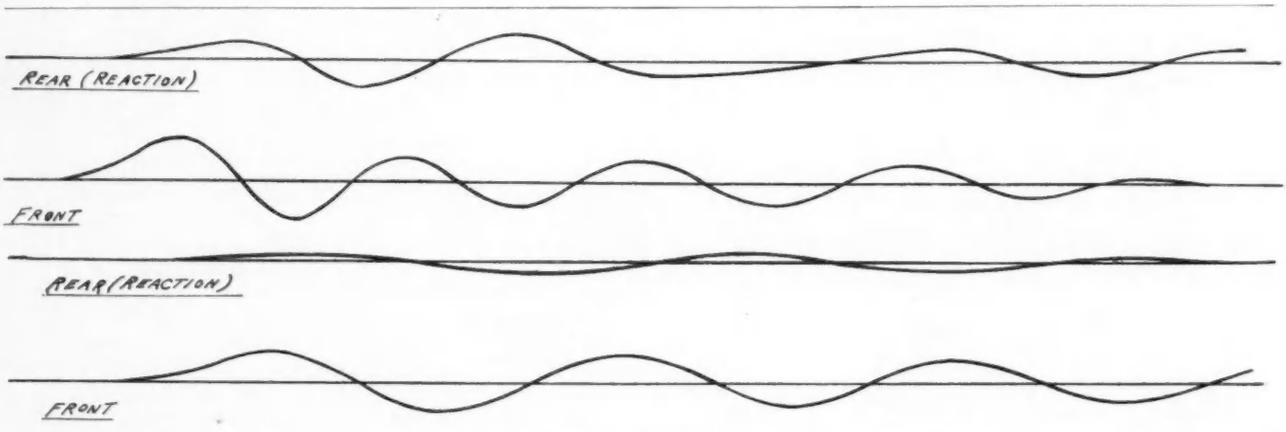

Fig. 9—Riding-Quality Model-Test Charts

These are typical charts obtained after front-wheel bump. For the upper front and rear curves the static deflection of the springs was, for the front, 2⅞ in. and for the rear, 5½ in. In this case $i^2 = 45.4$ sq. in., $AB = 56.25$ sq. in., and $i^2/AB = 0.81$

For the lower front and rear curves the static deflection of the springs was the same; but $i^2 = 50$ sq. in., $AB = 50$ sq. in., and $i^2/AB = 1.0$

Let us consider a car of conventional design. Assuming that the front wheels hit an obstacle, we find that the reaction on the rear springs will tend to move the body down in the rear. While this is occurring, the rear wheels have approached the obstacle and the body is displaced downward some more in addition to the initial movement due to the reaction, thus increasing the rebound. The unpleasant result is still aggravated because there is reaction on the front springs again from the rear wheels. This can be entirely overcome by the correct weight-distribution.

When a mass having considerable moment of inertia is suspended about a pivot point, and a force is applied to it at its center of gravity, a reacting force will also be acting on the pivot point in the same direction. This is due to the fact that the force has no lever arm about the center of gravity to overcome the moment of inertia of the mass. If the force is applied at a point away from the center of gravity, the reaction on the pivot will decrease. At last a point will be found where the force applied will give no reaction on the pivot point, as it will balance the inertia force of the mass. This point is called the center of percussion. If the point of force application is moved beyond the center of percussion, a reaction on the pivot point will result, only in the opposite direction. The pivot point and center of percussion are interchangeable.

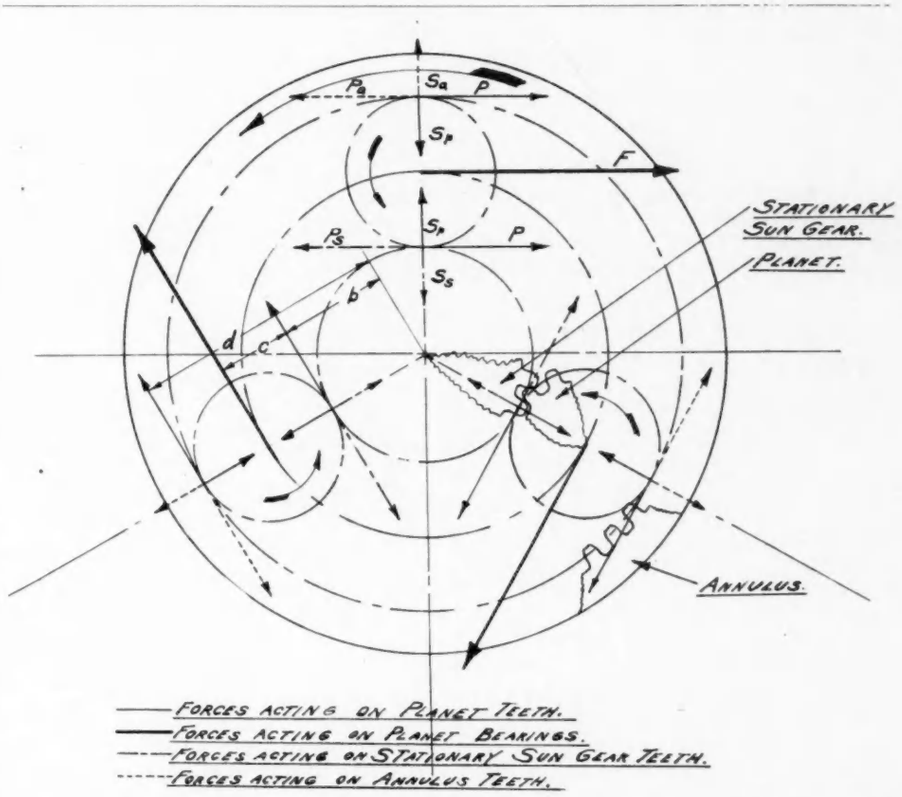

Fig. 10—Planetary System Which Acts When the Car Has Reached or Exceeded a Predetermined Speed

If, therefore, the center of percussion in a car is placed right at the rear axle when the front axle is the pivot point, and vice versa, complete independent action of both front and rear springs is obtained. To do this, a definite moment of inertia of the spring mass is required for a given wheel base and location of the center of gravity. This is the case when the two masses are concentrated over the wheel centers and the radius of gyration i is equal to the distance of the spring centers from the center of gravity.

In terms of an equation this means that, to obtain this ideal condition, $i^2 = AB$. The ratio i^2/AB is, therefore, an excellent indication as to the amount of interaction between front and rear springs. When $i^2/AB = 1$, there is no interaction, with the resulting good ride, and when it is either smaller or greater than unity, there will be interaction with the bad effects on the ride that follows, no matter what the spring rates and static deflections are.

In a car of conventional design, the ratio i^2/AB is approximately 0.75. Table 1 shows how the moving of the units shown in Figs. 6 and 7 increased or decreased the ratio i^2/AB. The Δ (i^2/AB), that is, increment of i^2/AB, was arrived at not only by considering the change in moment of inertia by reason of the different distance from the center of gravity of the car, but also by taking the change of weight of the different units into account.

The increase in moment of inertia of the body itself was not considered, as it would have been difficult to do that analytically. If all the factors could have been considered, it would have been found that the increase of i^2/AB would have brought the i^2/AB ratio of the airflow up to 1, as that is the value found by experiment. Table 1 is, therefore, not complete, but gives an indication of the importance of the relocation of the various units.

The relocation of the parts mentioned in Table 1 gave an increase of i^2/AB of 0.1597. Adding this to the original 0.75 of the conventional car, we get $i^2/AB = 0.9097$. The

Table 1—Showing How Moving the Units Shown in Figs. 6 and 7 Increased or Decreased the Ratio i^2/AB.

	Distance Moved. In.	Increase in Moment of Inertia, ΔI in Lb-Ft.[2]	Increase in i^2/AB. $\Delta(i^2/AB)$
Engine	18.0	8,874	0.1340
Radiator	18.4	969	0.0146
Steering Gear	34.9	84	0.0113
Front Bumper	7.5	600	0.0091
Rear Bumper	6.0	495	0.0075
Gasoline Tank	—0.3	7[a]	0.0001[a]
Battery	—11.9	—10	—0 0001
Spare Tire	—3.8	25[a]	0.0004[a]
Frame and Body (without increase in (I) of body)	—7.8	—1,173	—0.0172
		Total	0.1597

[a]Change in moment of inertia is positive in spite of less distance from the center of gravity because of greater weight of those parts in the airflow car.

Table 2—Maximum Car-Speeds and Reductions in Engine Speed

Axle Ratio	Overdrive Ratio	Maximum Car-Speed Without Overdrive: Car Speed, M.P.H.	Without Overdrive: Engine Speed, R.P.M.	With Overdrive: Car Speed, M.P.H.	With Overdrive: Engine Speed, R.P.M.	Reduction in Engine Speed with Overdrive at Maximum Car-Speed for the Ratio Used, R.P.M.
4.1	1.42	86.2	3,980	89.0	2,890	1,090
4.1	1.27	86.2	3,980	89.5	3,240	740
4.3	1.42	84.75	4,110	89.3	3,035	1,075
4.3	1.335	84.75	4,110	89.5	3,090	1,020

rest of 0.0903 missing to give the value 1 is mostly made up by the increased moment of inertia of the body itself, due to the fact that more weight is concentrated at the front and rear than in the conventional-type body.

In all airflow models, wind-tunnel tests showed a consistent gain where the width at the front of the body was greater than at the rear. Since the three-passenger rear-seat was desirable, and since the forward position of the rear seat gave the rear passengers greater comfort, the three-passenger front-seat resulted.

In this development, it has also been considered important to reclaim every possible capacity for passenger room and baggage space. Combined with this it was found that, due to the low floor and wide doors, the running board is less necessary as a step than as a protection to the body from road gravel and the like. A major consideration also is the body side-construction, which, due to the width of body, involves a bridge-type construction carrying directly from the arch over the front wheel to the sloping tail over the rear wheel. At first, it had been intended to make the front part of the radiator narrower, but it was found that in that way not enough cooling capacity could be obtained. The wide front was then

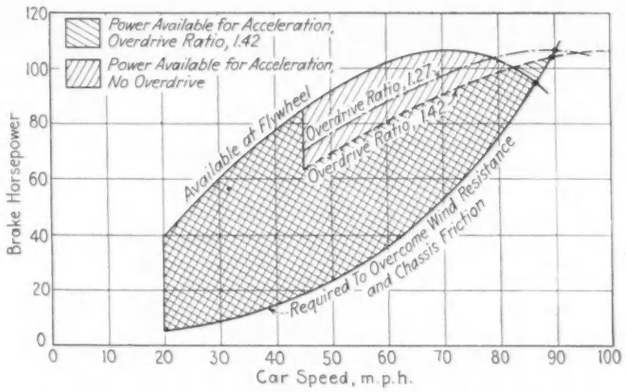

Fig. 12—Effects of the Overdrive, with 1.42 Ratio, on the Airflow Sedan having a 4.1 Axle-Ratio

adopted. The lower hood gives a much greater visibility, as most of the road can be seen over the front of the car.

The Overdrive as Part of the Airflow Car

In an effort to reduce the engine speed with the car traveling at high speeds, and to reduce gas and oil consumption and obtain longer engine life, the so-called "overdrive" was developed. The main object is to increase the useful range of the engine speed.

The overdrive consists of a clutch which brings a set of planetary gears into operation when the car has reached or exceeded a predetermined speed. The planetary system consists of a stationary sun-gear, planets running together on a carrier, and the annulus, as shown in Fig. 10. The forces acting on the gears—taken as spur gears in this case—and planet bearings are shown. With helical gears an additional thrust force acts on the planet bearing, which tends to increase the load on these somewhat. All other forces acting on the gear cancel themselves as can be seen from the diagram, re-

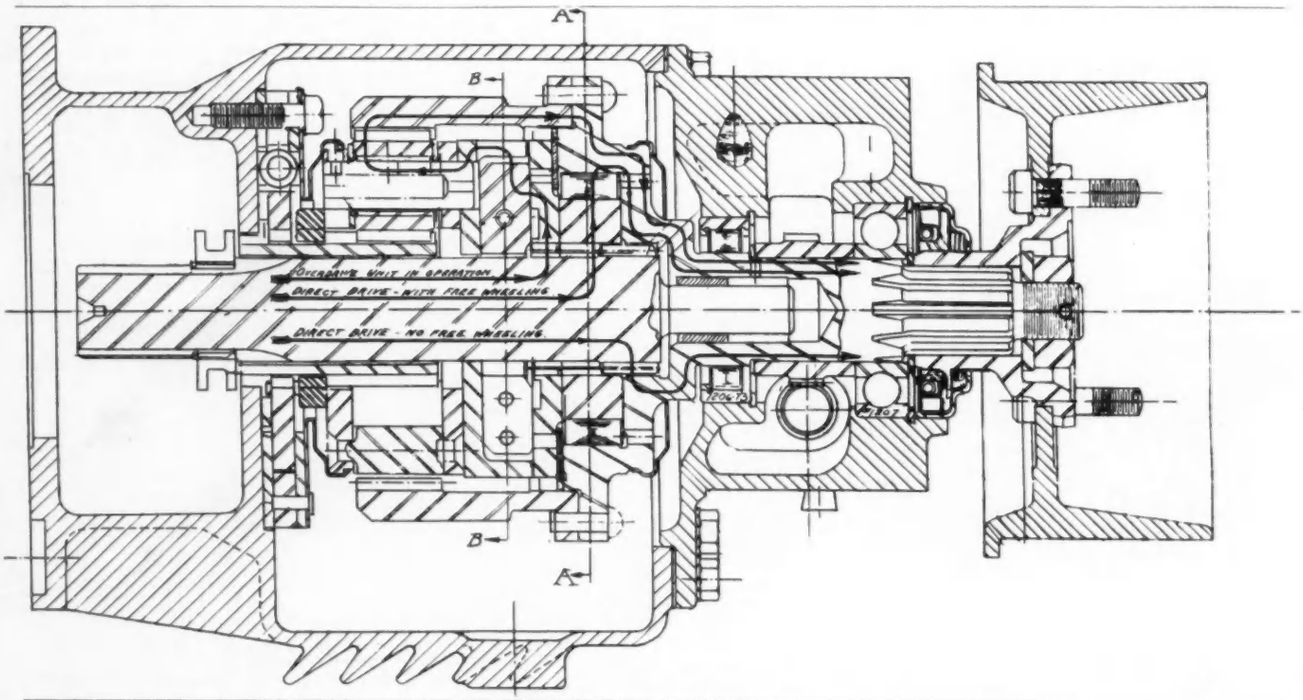

Fig. 11—Overdrive Arrangement

sulting in no eccentric loads. Fig. 11 shows the overdrive arrangement.

The ratio chosen for the overdrive on the airflow model is 1.42, and its effects on the performance of the car are shown in Figs. 12 and 13 for two different axle-ratios. The curves given are the ones for power required to overcome air resistance and chassis friction, and power available at the flywheel of the engine. The shaded areas are an indication of the power available for acceleration with and without overdrive. The maximum speeds are given by the intersections of the respective curves. The curves are all derived, and not based on actual tests.

With the 4.1 axle ratio, 2.8 m.p.h. top speed is gained with the 1.42 overdrive; and with the 4.3 axle, 4.55 m.p.h. is gained, but the maximum speed with the overdrive is practically the same for both axles.

An ideal overdrive ratio for consideration only of speed would be one that would bring the point of intersection of

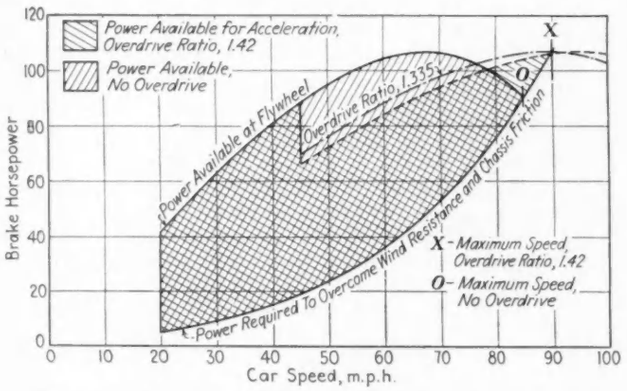

Fig. 13—Effects of the Overdrive, with 1.42 Ratio, on the Airflow Sedan Having a 4.3 Axle-Ratio

the two curves to occur at or slightly over the peak of the power-available curve. Such a ratio would result in less reduction of engine speed, but would increase the power available for acceleration. Since high-speed driving is only occasional, the best compromise is to reduce the engine speed as much as possible. Table 2 gives the maximum car speeds and reductions in engine speed for the two different axles with the actual overdrive ratio used, and the ideal. To obtain the ideal ratios, the planetary units would become excessive in diameter or compounding would be necessary.

Fig. 14 shows a typical analysis of power available, power required, and the drag composed of air resistance and chassis friction. In addition to showing the usual characteristics it shows that, with the overdrive, a greater car-speed can be reached under favorable wind conditions—exceeding the values shown on the curve—without an excessive engine-speed.

Appendix

Analysis of Forces Acting upon Planetary Overdrive Gears.—From Fig. 10 we have b/c turns of the planet for one turn of the carrier (the carrier carries the three planets); and c/d turns of the annulus per turn of the planet. Hence,

$$\left(\frac{b}{c}\times\frac{c}{d}\right)+1=\frac{b}{d}+1=\frac{b+d}{d}$$

turns of the annulus per turn of the carrier. The torque on the drive shaft is, therefore, $T_2=[d/(b+d)]\,T_1$, where T_1 is the torque delivered by the engine.

The tangential force at the annulus pitch-circle is:

$$F_T=\frac{T}{d}=\frac{d\times T_1}{d\,(b+d)}=\frac{T_1}{b+d}$$

The tangential force on the pitch circle of one planet is:

$$P=\frac{T_1}{(b+d)\times 3}$$

Separating Force between Annulus and Planet, and between Planet and Sun Gear.—We have $S_a=S_p=S_s=P\times\tan\alpha$, where α is the tooth-pressure angle. When helical gears are used in the system a thrust force is introduced, the magnitude of which is $T=P\times\tan\beta$, where β is the helix angle.

The force in the planet bearing is:

$$F=2P=\frac{2}{3}\times\frac{T_1}{b+d}$$

It can be seen from Fig. 10 that all the forces acting on the different gears cancel each other, except the tangential forces, P, on the planet which give a resultant load, F, on the planet bearing.

I wish to acknowledge great assistance from R. N. Janeway, Max Lassman and J. H. Kimpler, in preparing this review. They have done most of previous work to which I have referred.

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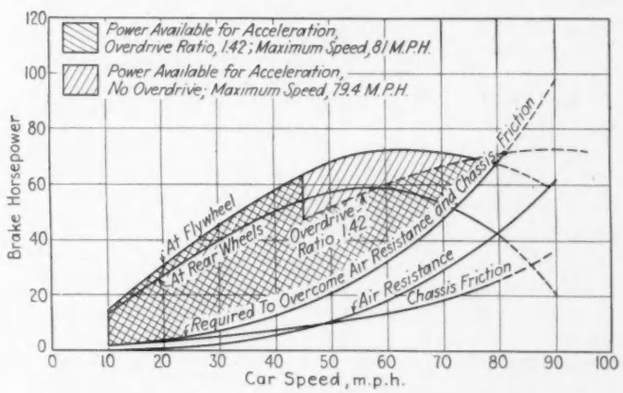

Fig. 14—Typical Analysis of Power Available, Power Required and the Drag Composed of Air Resistance and Chassis Friction, for the Airflow Sedan Having a 4.125 Axle-Ratio

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On Streamlining:

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A.T.Z—Automobiltechnische Zeitschrift, Feb. 25, 1934, p. 82; Wesen und Zweck der Stromlinienform.

Discussion of Rothrock Paper

Author's closure to discussion of the paper "A Photographic Study of Combustion in a Compression-Ignition Engine," by A. M. Rothrock, presented before the 1934 Annual Meeting and published in the S.A.E. JOURNAL, *June 1934.*

IN reply to Dr. Schweitzer's criticisms, I did not plot the ignition points as ignition lags because I feel that we are in danger of losing sight of the fact that the temperature and pressure of the air during the ignition lag are equally important as the time factor. In regard to the P_1/P_2 ratio, I stated that it was not a *direct* measure of combustion efficiency. It is nevertheless an approximate measure of combustion efficiency because the amount of fuel burned after the maximum pressure is reached is small enough to permit the ratio to be used for this analysis.

Under these circumstances I am still forced to wonder why the P_1/P_2 curves rise sharply with injection after top center. The same conclusion cannot be drawn with injection after top center as with injection before top center in the 20 to 30 deg. before top center range because, in the latter case, the combustion is accompanied by high rates of pressure rise and, in the former case, the rates of combustion are comparatively slow. Furthermore, and this is even more important, with the very early injection—greater than 40 deg. before top center—the combustion became progressively weaker. Again we see that the time element is not the whole story. Consequently, we must conclude that there are other factors to be considered than that of the amount of fuel accumulated in the cylinder before the start of burning. The full interpretation of the P_1/P_2 curves cannot be given simply on a time-lag or explosive basis.

Dr. Schweitzer is correct in stating in reference to Fig. 7 that the areas under the rates of heat exchange curves do not show a greater combustion efficiency with the longer ignition-lags. Any differences in these areas are well within the deviation necessitated by the method of computation. However, reference to the pressure curves does show that in each case the combustion efficiency at the higher temperature was the lower.

Dr. Schweitzer states that a short ignition-lag is beneficial from every standpoint, but he does not explain why. We have injected full-load fuel-quantity into a 5 x 7-in. engine in 0.0008 sec. The result was severe detonation, because, in my opinion, the injection period was less than the ignition lag. It is true that the lag might be shortened, but this cannot be done indefinitely because of certain chemical reactions which must take place before effective combustion is started. I do not think that shortening the injection period is necessarily what we want to do.

It is to me more reasonable to assume that a long ignition-lag will assure a good combustion-efficiency rather than a good cycle-efficiency. I do imply that injecting the fuel into the flame makes it more difficult to burn this fuel, because it has got to get out of this flame into fresh air containing oxygen. I have come to the conclusion that the vaporization is already sufficiently fast to obtain good mixing of the air and the fuel without the additional temperature of the flame, and it is of no use to increase the rate of vaporization if the rate of diffusion of the vapors is too slow to form a combustible mixture in the time available.

Well-built slow-speed engines cannot be compared to the high-speed engines because the injection period in the former is so much greater. Although the relative ignition-lags between the two engines may be the same, the actual ignition-lags in seconds is probably very different. I will grant that with sufficient time and temperature we can have combustion efficiencies of nearly 100 per cent. Whether or not the fuel in the slow-speed engine is introduced directly into the flaming gases will not be known until we have recorded these phenomena. The high-speed motion-pictures which we have already taken have given me a hearty respect for experimental data to substantiate such statements.

In conclusion, I think we must not put all our faith in shortening the ignition lag as a cure-all for increasing the performance of the high-speed compression-ignition engine. It is an important factor from some standpoints, but I believe the data presented in this report have shown that it also has inherent disadvantages, certainly in the quiescent combustion-chamber.

In regard to Mr. Marvin's discussion, I quite agree with him that there is a possibility that the combustion with detonation is of a different type from that without detonation. I am pleased that he has compared the N.A.C.A. results with those which have been obtained at the Bureau of Standards with the constant-pressure bomb. I believe that there is often a much closer relation between combustion with spark-ignition and combustion with compression-ignition than is generally realized. Particularly is this true of the chemistry of the combustion.

Low-Compression Spark-Ignition Oil-Burning-Engine Analysis

By Frederick H. Dutcher
Instructor, Columbia University

THE purpose of this paper is, first, to set down the methods which have been considered in the effort to develop a gasoline-type engine which will operate satisfactorily on non-volatile fuels, such as furnace or heating oil:

Second, to describe briefly commercial examples of the use of these various methods:

Third, to analyze the possibilities of these several processes with regard to power and efficiency, comparing them—as well as the results to be expected from the use of motor gasoline—with those of an engine operating on aviation gasoline as a basic standard.

Mr. Dutcher concludes that there are cases where horsepower hours per dollar's worth of fuel can be just as—or more—important than horsepower per pound of engine.

A further conclusion is that while the method used in the Appendix calculations is simple and the correction factors may be somewhat modified in the event that more accurate data should become available, it will be seen to give results which agree rather well with test records.

THE low-compression spark-ignition engine is not, as yet, a potent factor commercially. For years, efforts have been made to produce an engine that would develop comparable power on cheaper fuel than does the gasoline engine, but the results to date have not been such as to reduce appreciably the demand for gasoline.

Where fuel cost is of major importance, as in the truck, tractor and bus fields, and where fuel cost and fire hazard are almost equally important, as in the marine field, so long as there is any considerable price differential between light fuel-oil and gasoline this class of engine will be attractive commercially—if a cheap engine burning cheap fuel satisfactorily can be produced.

[This paper was presented at the March 15, 1934, Meeting of the Metropolitan Section, New York City.]

Any good spark-ignition engine must have incorporated in its design the following four basic functions:

(1.) Proper preparation of fuel and air before initiating combustion

(2.) Reliable ignition in a proper manner

(3.) Proper control of internal-metal temperature

(4.) Design characteristics which assure sufficient reliability, rigidity and facility for maintenance without frequent adjustment.

This paper is concerned with function (1), a proper condition of fuel and air before initiating combustion.

A proper mixture is one which is homogeneous and of the chemically combining proportions, while at the same time the temperature is as low as is consistent with dryness. An air-vapor-fog mixture in which all the fog is in suspension will have the same distribution and combustion characteristics as a dry mixture of air and vapor.

Homogeneity is required to assure first, equal distribution between the several cylinders of a multi-cylinder engine; second, that combustion shall be complete, for which reason also it is important that the chemically combining proportions be maintained. With regard to the lowest temperature which will permit dryness of the mixture, it is well known that, other things being equal, the mean effective pressure is proportional to the charge-weight per stroke. The temperature at the end of suction also limits the compression ratio that can be used, since the temperature at the end of compression must be so low as to assure that self-ignition of the charge shall not take place, and this—terminal-compression—temperature is dependent upon both the compression ratio and the temperature at the beginning of the compression—or end of the suction—stroke.

Since the primary reason for the use of a non-volatile fuel is to reduce fuel costs, it is important that the efficiency of the engine be as high as is consistent with smooth operation. Since efficiency is a function of the compression ratio, this ratio should be as high as possible, always provided that smoothness of operation be maintained.

It is a well-known fact that many gasoline carburetor-equipped engines give results, both as to power and economy, which cannot be considered good and which could be much improved were proper care taken to assure that equal charges of homogeneous mixtures, which would burn completely, were distributed to all the cylinders all the time. It is the function of the mixture-distributing system to distribute equal and uni-

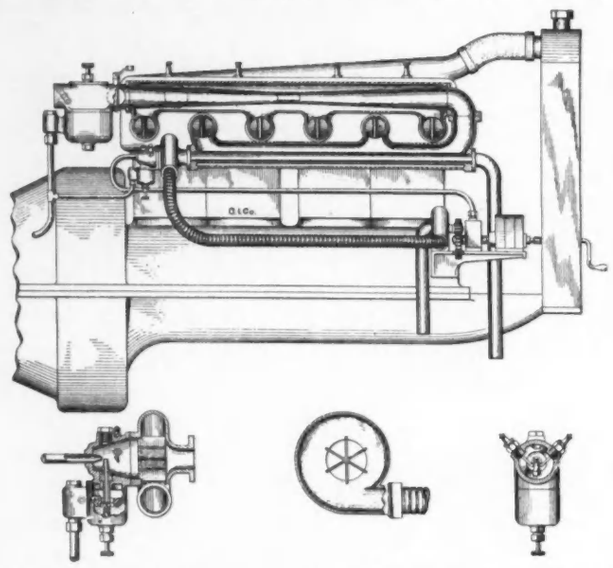

Fig. 1—Good Venturi Vaporizer, Equipped with a Blower

form charges of fuel to the several cylinders, and of the mixture-making system to provide a mixture that it is possible to so distribute. It is not possible for any distributing system to distribute *equally* a non-homogeneous mixture of vapor, fog, liquid-film and air throughout the range of speed and load over which any successful engine must operate, nor is it possible for such a mixture to burn completely after it is in the cylinder.

When the exhaust gases of an engine show simultaneously considerable percentages of both free oxygen and carbon monoxide, something is very wrong with the mixture-making system of that engine. Tests of gas engines show clearly that air-gas mixtures of the chemically combining proportions give best mean effective pressure with best thermal efficiency, and these with neither free oxygen nor carbon monoxide present in the exhaust gases.

There are three general methods of supplying low-volatility fuel-air mixtures, or fuel, to the class of engine under consideration, and the devices on the market and under development operate according to one or more of these methods:

(1) Preparation of an air-fuel-vapor mixture by evaporating the fuel in the presence of and mixing the vapor with all the air to be used

(2) Preparation of an air-fuel-vapor-fog mixture by making a primary vapor by heating the fuel in the presence of either no air at all, or a small amount of air, and subsequent addition of secondary air in sufficient quantity to bring the total air-fuel ratio within the operating range

(3) Direct injection of fuel into the combustion chamber, with ignition by spark

Many so-called oil-engines, built to operate according to Method (1), have been experimented with, and many of these have failed to take into consideration anything but the facts that the liquid fuel required heat to evaporate it and that an ample supply of heat was available in the exhaust gases. Consequently, the results were not particularly good because, in the effort to secure complete vaporization, the mixture was brought up to such high temperatures that detonation resulted under heavy loads. Further, at light loads and idling, either the amount of heat supplied, the area of vaporizing surface or the nature of that surface—individually or in combination—were so insufficient or ineffective that the resulting mixture was not only non-homogeneous, but contained a considerable amount of liquid. This caused smoke, carbon formation and fouling, and made operation unsatisfactory.

It was also necessary to use a volatile fuel for starting these engines as the "vaporizers" would not operate until they had been thoroughly heated, provision was rarely made for a start on anything but gasoline, the subsequent switch-over being made either manually or automatically as by some kind of thermostatic element.

The earlier examples of this type of vaporizer utilized a relatively small heating surface, which was maintained at high temperature. These generally produced a mixture not satisfactory for the entire load-speed range of the engine. An example of a typical device, which, however, was entirely satisfactory in operation, is the Good Venturi Vaporizer, Figs. 1 and 2, which was the subject of a paper[1] by Dr. Charles E. Lucke, read before the Society of Automobile Engineers in 1916. Fig. 1 shows one of these installations, which consisted of a long thin-walled venturi-shaped tube which extended the entire length of the engine, and was exposed not only to

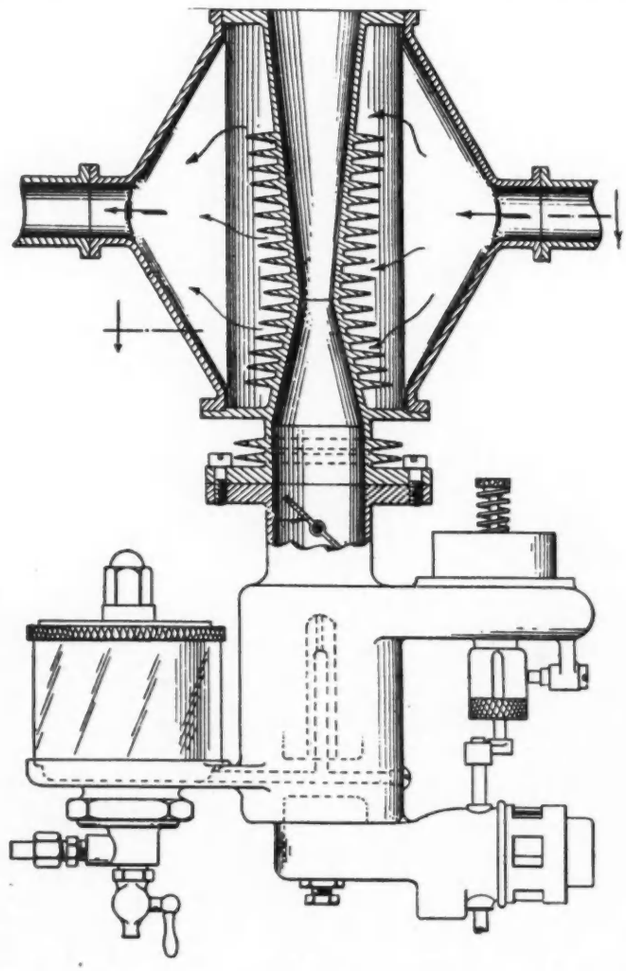

Fig. 2—Sectional View of the Good Venturi Vaporizer

[1] See S.A.E. TRANSACTIONS, vol. 11, part 2, p. 118; Kerosene versus Gasoline in Automobile Engines, by Dr. Charles E. Lucke.

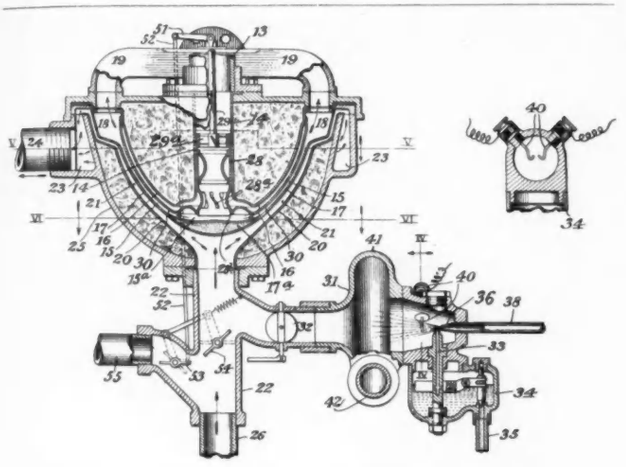

Fig. 3—Vaporizer Invented by John Good and Charles E. Lucke in 1920

heat by contact with the hot exhaust gases, but also to heat by radiation from the exhaust flames. Another form of this device, but which operated in the same general manner, is shown in Fig. 2. In this form the long horizontal tube was replaced by a shorter vertical mixing chamber, removed from the exhaust-port flames and finned on its outer surface, which was surrounded by an exhaust-heated jacket.

Unfortunately, test data are not available for these vaporizers. They are said, however, to have operated very satisfactorily on kerosene. Since present kerosene has an initial boiling-point of 369 deg. fahr. and an end-point of 485 deg. fahr.—a mean range of about 50 deg. fahr. below so-called furnace-oil, which is at present a favorite fuel for this class of engine—it is problematical whether this type of device could be considered as having commercial possibilities today. It does, however, illustrate a typical early example of vaporizer that was satisfactory and, as such, is included in this paper.

The general form of device using this same principle is any one which uses an extended surface on which to spread the liquid as a thin film, scrubbing this film with the air-stream, and supplying just enough heat to raise the temperature of the liquid to such a point that its vapor pressure is sufficient to allow complete vaporization.

Perhaps the best example of the application of this principle of partial-pressure evaporation is that shown in Fig. 3, a drawing made from the patent papers issued to Dr. Charles E. Lucke and John Good in 1920. It is believed that, since the claims granted for this patent so simply and completely state the principle involved, even though the patent is old and therefore presumably well known, a statement of the first two claims should be made herewith, as follows:

"*Claim 1.*—The method of making combustible gaseous mixtures consisting in combining air and liquid fuel in predetermined proportions suited for complete combustion of the fuel, depositing the liquid as a thin, extended film upon a heated surface, and coincidently forcibly removing the vapor from such heated liquid film by advancing the air in the form of a stream of high velocity over and in frictional contact with the surface thereof while maintaining the temperature of said film lower than that which decomposes the liquid."

[2] See S.A.E. TRANSACTIONS, vol. 23, p. 358, Dew-Point Data on Gasoline and vol. 24, p. 240, Present Status of Equilibrium-Volatility Work at the Bureau of Standards, by Oscar C. Bridgeman.

"*Claim 2.*—The method of making combustible gaseous mixtures consisting in combining air and liquid fuel in predetermined proportions, depositing the liquid upon an extended heated surface of progressively increasing area, passing the said air as a shallow expanding stream over said surface with sufficiently high velocity to advance the liquid as a thin, spreading film on said surface, and coincidently changing the direction of the said high velocity shallow stream."

An example of the commercial use of this principle is shown in Fig. 4, the Godward Vaporizer, which was produced in this country some years ago and which was rather extensively investigated at that time. These Godward vaporizers were originally developed for use with gasoline, and Fig. 5 shows comparative power and efficiency curves for an engine using gasoline in its standard mixture-making and distributing system and in the Godward system of manifolds. It will be noted that the results with the Godward system, which produced a dry mixture, are much better than those produced by the other manifolds. This indicates a line of investigation that might well be followed by any manufacturer who is really anxious to improve economy with no sacrifice of power, and, better still, with an improvement in power. The performance of this device with fuels of low volatility was not, however, such as to make it a success, and its manufacture was discontinued. The results for operation on gasoline and on fuel oil are shown in Fig. 6.

Another vaporizing device of the hot-mixture type is the Paragon, a section of which is shown in Fig. 7. This device is ingenious in that it combines the hot-mixture method—for all but full-load positions of the throttle—with the air-vapor-fog mixture type for wide-open throttle. In this way, a dry mixture is provided at partial throttles, where a reduced charge-density is required; while the density-reducing effect of high temperature is lessened at full throttle, when maximum power is desired.

Test curves from an engine operating both on gasoline with its standard equipment and on fuel oil with Paragon-Vaporizer equipment are shown in Fig. 8. The equilibrium air-distillation work done at the Bureau of Standards[2] by O. C. Bridgeman furnishes methods of calculating the mini-

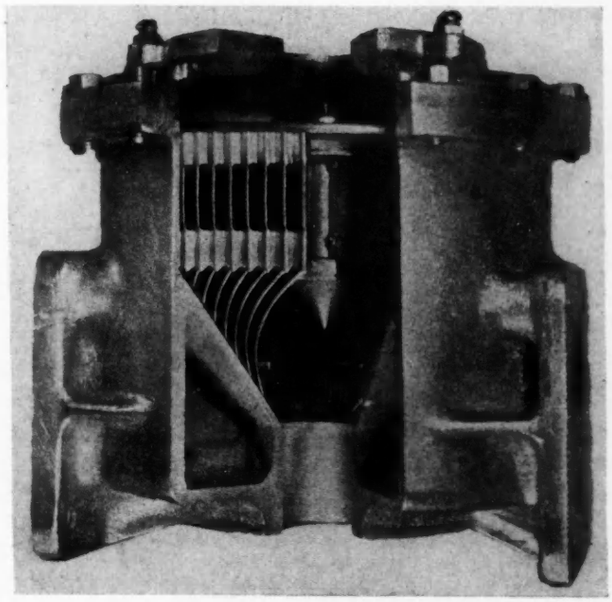

Fig. 4—The Godward Vaporizer

mum temperatures required for dry mixtures of hydrocarbon fuels and air. With these data and a knowledge of the exhaust-gas temperatures available, it should be possible for a designer knowing something about heat transfer to calculate fairly closely the vaporizer surface-area required for any given engine of known specifications, operating characteristics and desired fuel.

Method (2) of mixture preparation is exemplified by those devices which, by the application of considerable heat to the fuel and a relatively small amount of air or no air at all, form a very rich and hot mixture or a hot fuel-vapor, which is then cooled by the admixture of cool secondary air in sufficient quantity to bring the total air-fuel ratio to the chemically combining proportions. It is known that the fog resulting from such a mixture of hot vapor and cool air will be homogeneous, and will maintain its fog condition for a considerable time. It is also of interest that the fog is finer the heavier the oil and the cooler the air.

These devices would seem to have better possibilities both as to power and efficiency, than the first or "hot-mixture" type, since the fog which is formed upon the mixing of the cool air and hot-mixture streams will persist for a considerable period of time and has approximately the same distribution and combustion characteristics as an air-vapor mixture. As previously noted, the Paragon vaporizer operates in this manner at the wide-open-throttle position.

Fig. 9 shows another device in this classification, the Thurberator, which, under all conditions of throttle opening, operates as a fog-mixture maker. As shown, the carburetor which supplies the rich mixture has an exhaust-heated float-chamber and is also supplied with hot air from an exhaust-heated stove. Mixing with the cool secondary air, which, like the primary air is metered by a spring-loaded valve, is effected just as the primary and secondary streams enter the manifold.

Fig. 10 shows comparative performance results from a six-cylinder Knight-type bus-engine which indicate superior per-

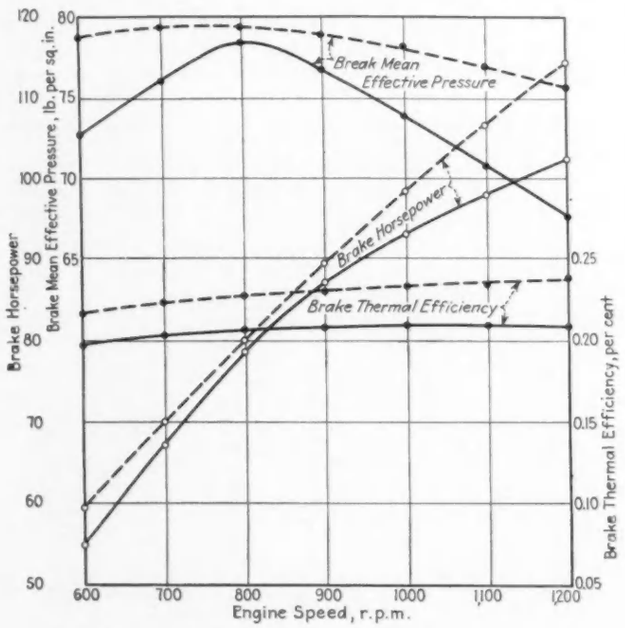

Fig. 5—Performance Curves of a Six-Cylinder 5½ x 7-In. Gasoline-Engine, Using Gasoline Fuel and a Standard Compression Head

The solid-line curves are for standard equipment; the dash-line curves, for Godward-Vaporizer equipment.

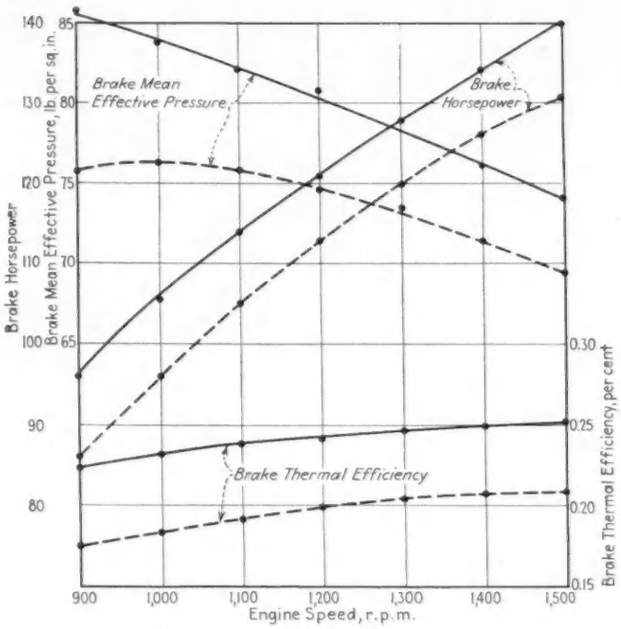

Fig. 6—Performance Curves of a Six-Cylinder 5½ x 7-In. Gasoline-Engine, Using Gasoline and Using Fuel Oil as Fuel, with High Compression

The solid-line curves are for gasoline; the dash-line curves, for fuel oil.

formance with the non-volatile fuel, properly treated, as compared with standard equipment and gasoline. It would be interesting to see just how good a performance one of these re-vamped engines could give on gasoline if the same pains were taken with the mixture-making and distributing system as are evident in the rest of the design.

It is perhaps necessary to point out now that the most interesting thing about the comparative results noted here and elsewhere is not the excellent performance of the particular device under consideration, but, rather, the uniformly poor results, comparatively, of "standard" equipment.

Method (3)—that of direct injection of fuel into the cylinder—is by no means the least important of the three methods of charge preparation in this line. This is exemplified in present commercial practice by the Waukesha-Hesselman Type-HAH engine, test results from which are shown in Fig. 11. From the purely academic viewpoint, this type has perhaps the best power and best economy-potentialties of the three, since it is unnecessary to heat the incoming charge at all and plenty of fuel pumps are available that will give accurate metering and timing of the oil charge. But from the practical angle, some difficulties can be seen. Unless the fuel is completely mixed with the air charge—in other words, unless the air-fuel mixture in the combustion chamber is completely homogeneous—there will be regions within the combustion chamber where the mixture is of other than the combining proportions, in which case the combustion efficiency of the engine will be less than perfect. It is also possible that should an area of non-combustible *mixture* occur, so placed as to check flame travel, a considerable portion of the charge may remain unburned. If this condition obtains, the indicated mean effective pressure will of necessity be lower than the optimum, as will indicated thermal efficiency also, and power and economy will cease to be competitive with the power and economy of a gasoline engine.

The preceding text has defined what a "proper" mixture is,

and given a short description of the various methods by which such a mixture can be produced. An attempt will now be made, by means of logical assumptions combined with existing factual data, to analyze the possibilities of an engine supplied with proper mixtures made by Methods (1) to (3) inclusive.

The four fuels to be used are: Aviation gasoline, motor gasoline, furnace oil and heating oil. The fractionation curves for these fuels are shown in Fig. 12.

According to the work of Bridgeman[3], the minimum dry-mixture temperature for a mixture of hydrocarbon fuel and air is determined from the following equations, the temperatures being in degrees Fahrenheit, absolute.

$$\frac{T_{90\ ASTM}}{T_{DP}} = 1.084 + 0.185 \log \left[(1+3.9M)\left(\frac{760}{P}\right)\right] \qquad (1)$$

or

$$T_{DP} = \frac{T_{90\ ASTM}}{1.084 + 0.185 \log \left[(1+3.9M)\left(\frac{760}{P}\right)\right]} \qquad (2)$$

where

$T_{90\ ASTM}$ = Temperature at the A.S.T.M. 90-per cent point.
T_{DP} = Temperature at the dew point.
M = Air-fuel ratio, by weight.
P = Absolute total mixture pressure, in millimeters of mercury.

[3] See S.A.E. TRANSACTIONS, vol. 23, p. 358, Dew-Point Data on Gasoline and vol. 24, p. 240, Present Status of Equilibrium-Volatility Work at the Bureau of Standards, by Oscar C. Bridgeman.

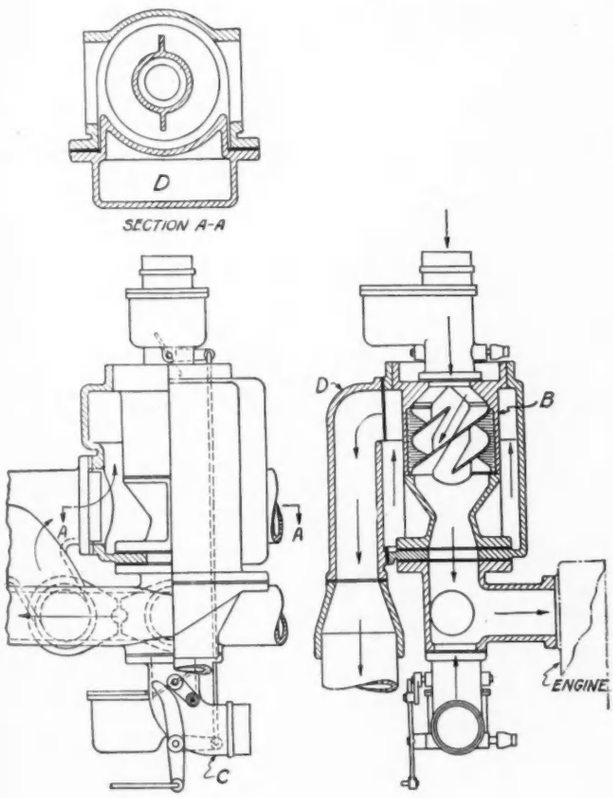

Fig. 7—The Paragon Vaporizer

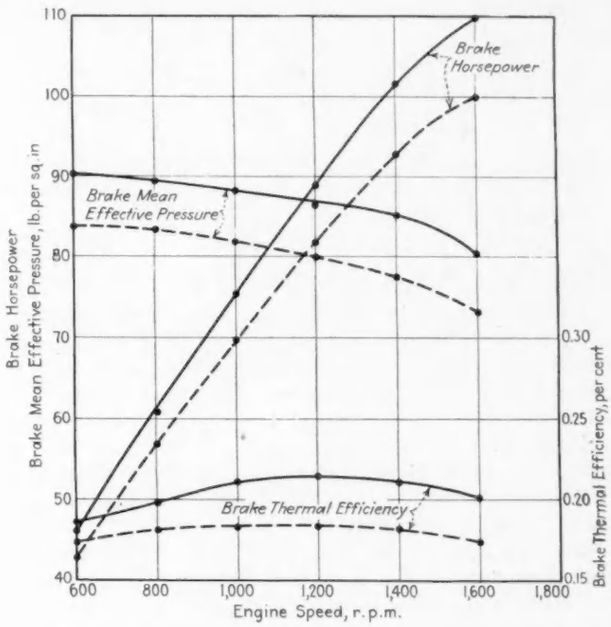

Fig. 8—Performance Curves of a Six-Cylinder 5 x 5¾-In. Engine, Using Gasoline Fuel with Standard Equipment and Distillate Fuel with the Paragon Vaporizer

The solid-line curves are for gasoline; the dash-line curves, for distillate.

Substituting the proper values in Equation (2) and calculating for an air-fuel ratio of 15, which is close enough to the combining proportion for most fuels, the minimum dry-mixture temperatures can be determined.

Since all fuels, when mixed with air, have approximately the same number of British thermal units per cubic foot of chemically correct mixture—namely, ± 105 B.t.u. at standard conditions of a temperature of 32 deg. fahr. and a pressure of 14.7 lb. per sq. in.—and provided that air-fuel mixtures can exist under these conditions, no great error will be caused in the calculation if this value be used for all the fuels under consideration. The density and heat-content ratios for the several air-fuel mixtures will be proportional to the absolute temperatures of the dry mixtures at standard pressure.

It will be seen presently that the minimum dry air-vapor-mixture temperatures for the oils commonly used in the type of vaporizer under discussion are so high that the power available is considerably less than can be obtained with gasoline. Further, due to these high temperatures on the one hand and to the characteristics of the fuel on the other, detonation is severe with any considerable compression ratio. The next step, therefore, is the production of a fog-mixture by means of mixing a hot vapor with unheated air in sufficient quantity to give an operating air-fuel ratio.

The limiting condition for this is that obtaining when the oil is vaporized without admixture of air, and the vapor condensed to a suspended fog upon mixing with the cold air-stream. The temperature of the vapor as produced is very close to that of the mean ordinate of the fractionation curve and, since the latent heats of the fuels used run about 150 B.t.u. per lb., the temperature rise of the secondary air is due to the heat of condensation of the fuel vapor, as in Equation (3).

$$Wt._A(T_m - T_A)(C_pA) = Wt._o \times LH_o + Wt._o(T_v - T_m)(C_pO) \quad (3)$$

where

$Wt._A$ = Weight of air.

T_m = Mixture temperature.

T_A = Air temperature.

C_pA = Specific heat of air at constant pressure = 0.24.

$Wt._o$ = Weight of oil.

LH_o = Latent heat of oil.

T_v = Vapor temperature.

C_pO = Specific heat of oil at constant pressure = 0.50.

The air-fuel ratio for Equation (3) is 15. This equation is based on the hypothesis that all the fuel vapor is condensed and all the condensate is in the form of fog. This condition cannot obtain, since a portion of the vapor will not condense but will remain as vapor at very low partial pressure and density. I am indebted to J. L. R. Morgan, Professor of Physical Chemistry at Columbia University, for the suggestion that the weight of vapor in a mixture of this sort will hardly exceed 10 per cent.

Referring again to the work of Bridgeman[4], the temperatures of mixtures in the form of 10 per cent of vapor and 90 per cent of fog, for the furnace oil and the heating oil under discussion, will be, respectively, 640 and 662 deg. fahr. absolute. Calculations for these 10/90 vapor-fog mixtures are shown, as well as those for the hypothetical 100-per cent fog-air mixtures, in both Table 1 and Table 2. Calculations are also shown, in Table 2, for 60-per cent air-utilization with injection for the two non-volatile fuels.

Regarding the injection process, the amount of fuel which can be utilized per cycle depends upon (*a*) the weight of air into which the fuel is injected, and (*b*) the ability of the fuel to find air with which to combine. In the absence of data to the contrary, there seems to be no good reason to suppose that the air utilization of an injection-oil engine with spark ignition will be any better than that of an injection engine with compression ignition; namely, about 60 per cent. However, since it is the purpose of this paper to predict the power obtained by the perfect operation of the three processes under discussion, the injection process will be treated for both 100-

[4] See S.A.E. TRANSACTIONS, vol. 23, p. 358, Dew-Point Data on Gasoline and vol. 24, p. 240, Present Status of Equilibrium-Volatility Work at the Bureau of Standards, by Oscar C. Bridgeman.

Table 1—Fog-Air and Vapor-Fog Mixture-Data

Fuel	Minimum Dry Mixture: Air-Vapor Temp. Deg. Fahr. Abs.	Minimum Dry Mixture: Air-Vapor Temp. Deg. Fahr.	Density Ratio Based on Aviation Gasoline	B.T.U. per Cu. Ft.
Aviation Gasoline..	510	50	1.000	101.2
Motor Gasoline...	575	115	0.886	89.9
Furnace Oil.....	690	230	0.739	74.9
Heating Oil.....	725	265	0.703	71.3
100-Per Cent Fog-Air Mixture				
Furnace Oil.....	603	143	0.846	86.8
Heating Oil.....	606	146	0.842	85.4
10-Per Cent Vapor, 90-Per Cent Fog-Air Mixture				
Furnace Oil.....	640	180	0.798	81.0
Heating Oil.....	662	202	0.771	78.2

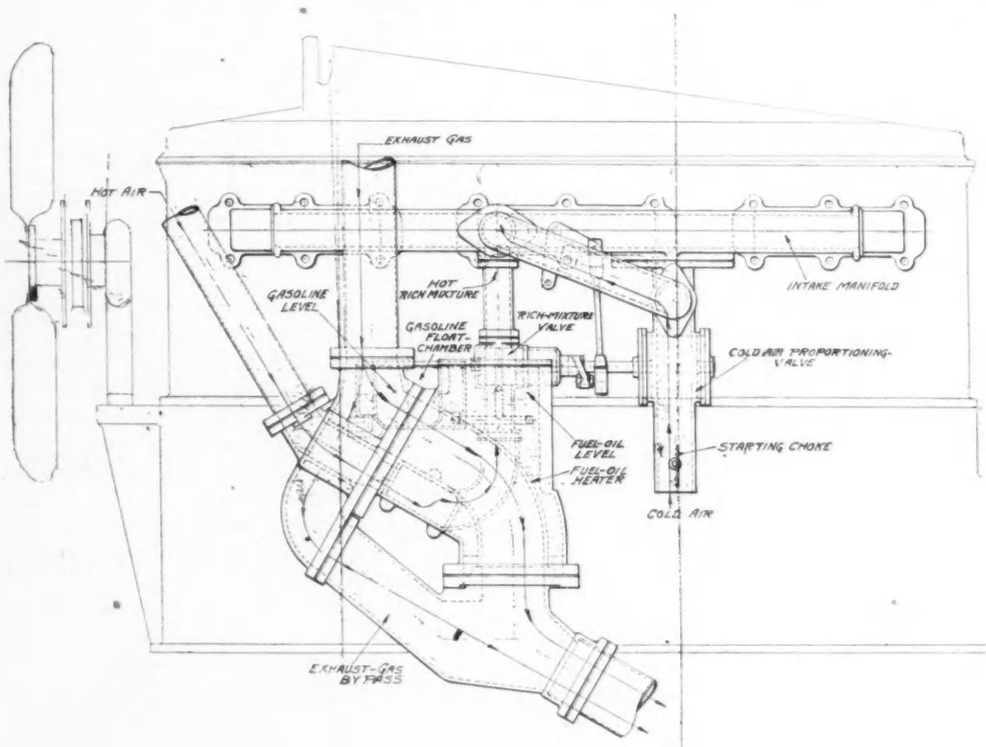

Fig. 9—The Thurberator, Which Operates as a Fog-Mixture Maker under All Conditions of Throttle Opening

per cent and 60-per cent air-utilization. The higher of these values is an academic possibility, but hardly a mechanical probability.

It now remains to set up a rational method for predicting the relative mean effective pressures that can be realized as maxima by Methods (1) to (3) inclusive, using the four fuels specified. It is obvious that such a prediction cannot be expected to give absolutely accurate results. For any given engine, however, the relative predicted values may be expected to be close enough to those actually obtainable in operation as to justify the method of calculation.

The ideal values of mean effective pressure (*M. E. P.*) and thermal efficiency E_t for the Otto cycle are represented by Equations (4) and (5), as follows:

$$E_t = 1 - (1/R_v^{0.408}) \tag{4}$$

$$M.E.P. = (W/144R_v) = \frac{JQ}{144V_a}\left[\frac{1-\left(\frac{P_a}{P_b}\right)^{[(s-1)/s]}}{1-\left(\frac{P_a}{P_b}\right)^{1/s}}\right] \tag{5}$$

where

W = Work, in foot pounds per cycle.
R_v = Ratio of volumes.
J = Joule's equivalent = 778.
Q = Heat added, in British thermal units per cycle.
V_a = Volume (clearance + displacement).
V_b = Clearance volume.
P_a = Pressure at point a (indicator card).
P_b = Pressure at point b (indicator card).
s = 1.41 = (C_{pA}/C_{vA}).
C_{pA} = Specific heat of air at constant pressure.
C_{vA} = Specific heat of air at constant volume.

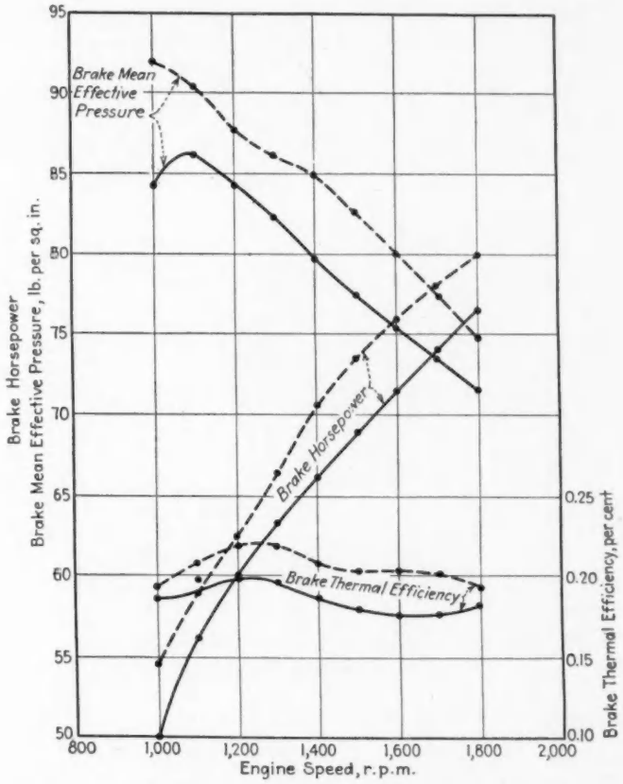

Fig. 10—Performance Curves of a Knight-Type Six-Cylinder 4¼ x 5½-In. Gasoline- Engine, Using Gasoline with Standard Equipment and Fuel Oil with the Thurberator

The solid-line curves are for gasoline; the dash-line curves, for fuel oil.

Table 2—Data for the Hypothetical Indicator-Cards under the Various Conditions of Fuels and Mixture Preparation[a].

	Dry (Air/Vapor) Mixtures				Fog Mixtures: Air/Fog				Injection 100-Per Cent Air-Utilization		Injection 60-Per Cent Air-Utilization	
							Air-Vapor-Fog					
	Aviation Gasoline	Motor Gasoline	Furnace Oil	Heating Oil	Furnace Oil	Heating Oil	Furnace Oil	Heating Oil	Furnace Oil	Heating Oil	Furnace Oil	Heating Oil
P_A	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7
T_A	510	575	690	725	603	606	640	662	520	520	520	520
P_B	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0
T_B	552	617	732	767	645	648	682	704	562	562	562	562
P_C	133	133	133	133	133	133	133	133	133	133	133	133
T_C	965	1,078	1,280	1,340	1,125	1,130	1,190	1,228	1,065	1,065	1,065	1,065
Ratio (K)[b]	1.000	0.886	0.739	0.703	0.864	0.842	0.798	0.771	0.9375	0.9375	0.563	0.563
Q	978	865	722	687	828	825	781	755	916	916	550	550
C_v	0.3427	0.3395	0.3480	0.3385	0.3321	0.3320	0.3420	0.3405	0.3440	0.3440	0.3140	0.3140
T_D	3,815	3,623	3,358	3,368	3,620	3,615	3,472	3,445	3,725	3,725	2,815	2,815
P_D	526.0	447.4	348.0	334.5	431.5	426.0	388.0	374.0	465.0	465.0	351.0	351.0
"s" (Expansion)	1.17	1.18	1.20	1.20	1.18	1.18	1.19	1.20	1.17	1.17	1.225	1.225
P_E	72.8	61.0	45.8	44.0	58.7	58.0	51.9	49.2	64.4	64.4	44.1	44.1
T_E	2,860	2,675	2,370	2,400	2,670	2,665	2,520	2,455	2,795	2,795	1,992	1,922
Mean Exp. Pressure	174.5	147.0	113.0	108.2	141.9	139.9	127.0	121.3	154.5	154.5	112.0	112.0
Mean Comp. Pressure	39.5	39.5	39.5	39.5	39.5	39.5	39.5	39.5	39.5	39.5	39.5	39.5
Gross I. M. E. P.	135.0	107.5	73.5	68.7	102.4	100.4	87.5	81.8	115.0	115.0	72.5	72.5
Losses M. E. P.	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0
Brake M. E. P.	119.0	91.5	57.5	52.7	86.4	84.4	71.5	65.8	99.0	99.0	56.5	56.5
Ratio, Exp. M. E. P.	1	0.843	0.649	0.62	0.813	0.799	0.729	0.695	0.885	0.885	0.641	0.641
Ratio, Gross I. M. E. P.	1	0.796	0.545	0.51	0.76	0.743	0.649	0.605	0.851	0.851	0.536	0.536
Ratio, Brake M. E. P.	1	0.770	0.483	0.44	0.726	0.710	{0.553 / 0.601}	0.553	0.832	0.832	0.475	0.475

[a] See Fig. 13 for designated points *A*, *B*, *C*, *D* and *E*, on the indicator card.
[b] Ratio K = Charge density at $P = 14.7$ lb. per sq. in. and T at the manifold entrance to a dry mixture of aviation gasoline and air; or, the ratio in British thermal units per specific volume at the same pressure, P.

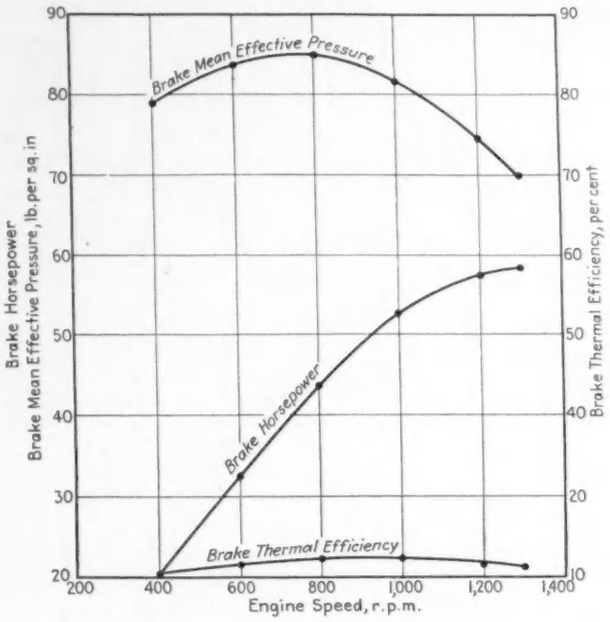

Fig. 11—Performance Curves of a Four-Cylinder 5 x 6½-In. Waukesha-Hesselman Type-HAH Engine

Calculations will be facilitated if Equation (5) is converted into one expressed in volume rather than pressure relations, as in Equation (6):

$$M.E.P. = \frac{JQ}{144 V_a}\left[\frac{1-\left(\frac{V_b}{V_a}\right)^{\epsilon-1}}{1-\left(\frac{V_b}{V_a}\right)}\right] \qquad (6)$$

[5] See National Advisory Committee for Aeronautics Report No. 102; Performance of a Liberty-12 Airplane Engine, by Sparrow and White.

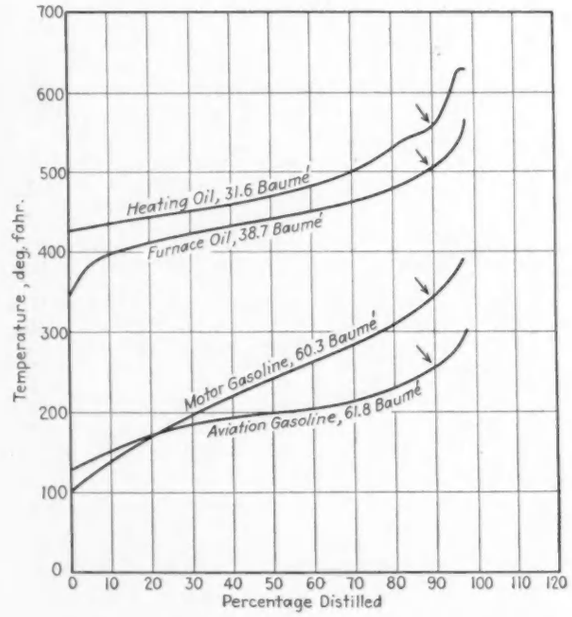

Fig. 12—Distillation Curves of Four Fuels Used in Analyzing the Possibilities of an Engine Supplied with Proper Mixtures Made by Methods (1) to (3) Inclusive

Substituting for standard conditions $Q = 105$ B.t.u. per cu. ft. and $V_a - V_b = 1$, we have: $M.E.P. = 348$ lb. per sq. in.

The foregoing equations represent the ideal possibilities of the Otto cycle under the conditions of pure air with constant specific heats at a constant volume of 0.17 and at a constant pressure of 0.24. These conditions do not, of course, obtain, since (*a*) the working fluid is not air, but a mixture of air and fuel which undergoes chemical change, and (*b*) the specific heats of the several gases which make up the products of combustion are not constant, nor do they vary in like degree.

To determine approximately what the actual efficiency and mean effective pressure due to any given condition of compression ratio and air-fuel mixture are, the simplest procedure is to construct an hypothetical indicator-card based on accepted values for variable specific heats of gases, and the simple pressure-volume-temperature relations for gases. To

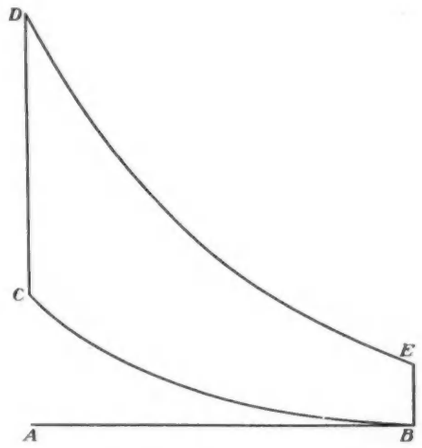

Fig. 13—Indicator Card on Which the Calculations Are Based

have a check on the corrections of the calculations, an engine was chosen on which many data were available, that is, the Liberty airplane-engine, with 5.42 compression ratio, which produced, according to test reports[5] a brake mean effective pressure of 117 lb. per sq. in. and which had a "losses mean pressure" (friction and pumping) of 16 lb. per sq. in. at 1700 r.p.m. with a volumetric efficiency of 85 per cent and an air-fuel ratio of 14. The point-by-point construction of the hypothetical indicator-cards for this engine under the various conditions of fuels and mixture preparation appear in Table 2. A complete set of sample calculations for the aviation-gasoline dry-mixture card appears in the appendix to this paper, but it seems unnecessary to go into this matter in detail at this time.

It will be noted that the ratios of the respective indicated mean effective pressures for the several conditions are somewhat lower than are the corresponding ratios of the several charge densities. This is explained by the fact that the gross indicated mean effective pressure is the algebraic sum of mean expansion pressure (a variable) and mean compression pressure (a constant). The gross indicated mean effective pressure equals the mean expansion pressure, minus the mean compression pressure. For a given compression ratio and terminal suction pressure, the mean compression

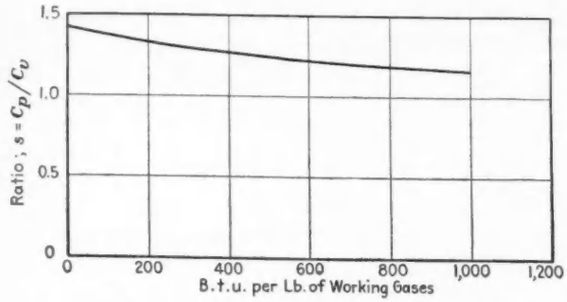

Fig. 14—Relation between the Ratio of Specific Heats at Constant Pressure and Constant Volume, C_p/C_v, and the Number of British Thermal Units per Pound of Working Gases, as Calculated from the Data of H. R. Ricardo[10]

pressure is constant. The mean expansion pressure, however, is proportional to the height of the explosion (or combustion) line, which, in turn, depends upon the heat added per cycle, or per pound of working gases, the mean specific heats of the products of combustion and the radiation losses.

Since it is perfectly evident that X/Y is not equal to $X—Z/Y—Z$, it is obvious that the ratios of the indicated mean effective pressures obtainable with the different mixtures under consideration will not be the same as the ratios of the respective charge weights per cycle. Likewise, since we can assume that the pumping and friction losses of the engine will be the same for the general cases, the same reasoning accounts for the still greater variation in the ratios of the brake mean effective pressures.

It is evident that it is not logical to expect as high power from an engine designed for satisfactory operation on gasoline when a less-volatile fuel is used; that if it were possible to obtain complete homogeneity of the air-fuel charge, the injection type of low-compression spark-ignition oil-burning engine would deliver power more nearly approaching that obtained from gasoline than do the other methods; and that the performance with fog mixtures, while potentially not so good as that by the injection method, is considerably superior to that of wholly dry vapor-air mixtures.

These conclusions bear directly upon the power which can be developed from a given engine with a constant compression-ratio, operating on several different fuels. It seems in order to suggest that, for certain classes of service, there are other things to be considered than the maximum brake mean effective pressure that can be developed by a given machine.

We know that, other things being equal, the indicated thermal efficiency depends upon the compression ratio. We know also that "other things" are usually not equal.

Referring to Tizard and Pye's figures[6], which give the efficiency equation as $E = 1 - (1/R_v^{0.258})$, we see that the value of E, for $R_v = 5$, is 0.34; while the value of E, for $R_v = 4$, is 0.30. Hence there is a difference of 4 per cent in favor of the higher compression-ratio, and this is just about the difference in efficiency due to 1 per cent of carbon monoxide in the exhaust. It indicates that, in the development of a cheap engine to burn cheap fuel cheaply, an engine may be just as good—or better—for a particular job, even though it may weigh a few pounds more per horsepower than does another engine which gets its higher power through wasteful use of fuel. In other words, there are cases where horsepower-hours per dollar's worth of fuel can be just as—or more—important than horsepower per pound of engine.

[6] See The Internal-Combustion Engine, by D. R. Pye; Oxford University Press, London, 1931.
[7] See National Advisory Committee for Aeronautics Report No. 102; Performance of a Liberty-12 Airplane Engine, by Sparrow and White.
[8] See *Automotive Industries*, Sept. 23, 1926, p. 488; A Study of a Stock Engine with an Indicator, by S. R. Treves.
[9] See National Advisory Committee for Aeronautics Report No. 102; Performance of a Liberty-12 Airplane Engine, by Sparrow and White.
[10] See *The Automobile Engineer* (London), vol. 11, June, 1921, p. 219; Some Experiments on Supercharging a High-Speed Engine, by H. R. Ricardo.

Appendix

Calculation of the indicator card shown in Fig. 13 is presented in the subsequent text. The card is for a Liberty-12 airplane-engine[7] having characteristcs and operating conditions as follows:

Bore, in.	5
Stroke, in.	7
Compression ratio (by volume)	5.42
Engine speed, r.p.m.	1,700
Brake mean effective pressure, lb. per sq. in.	117
Losses in mean effective pressure (friction and pumping), lb. per. sq. in.	16
Gross indicated mean effective pressure, lb. per sq. in.	133
Volumetric efficiency (air), per cent	0.85
Air-fuel ratio	15
Fuel	Aviation Gasoline

The points *A, B, C, D* and *E*, shown in Fig. 13, are used as subscripts to the respective symbols for pressure, volume and temperature; that is, P_A, V_B, T_C and the like.

A—Charge Temperature (T S T) at the Beginning of Compression:

Examination of specimen indicator cards for engines of this general type[8] indicates that, due to charge velocity in the manifold and through the valves—that is, "ramming effect"—the apparent E_v is approximately 0.98 per cent. It is noted in test reports of this engine[9] that E_v was calculated from air measurements. Obviously, if a 15 : 1 air-fuel ratio is used, for 15 lb. of air entering the carburetors 16 lb. of mixture will enter the engine. It is therefore necessary to correct the recorded values, for all but the injection calculations, by 16/15.

The reported value for E_v is 0.85 per cent. Corrected, the real $E_v = 0.85 \times 16/15 = 0.906$. The E_v is inversely proportional to suction resistance and suction heating; that is:

$$E_{v\ real} = E_{v\ apparent}\left(\frac{T_{mixture}}{TST}\right) = 0.906$$

$$= 0.980\left(\frac{510}{TST}\right)$$

$$TST = T_{mixture}\left(\frac{E_{v\ apparent}}{E_{v\ real}}\right)$$

$$= 510\left(\frac{0.980}{0.906}\right) = 552 \text{ deg. fahr.}$$

$$SH = TST - T_{mixture} = 552 - 510 = 42 \text{ deg. fahr}$$

The assumption is made that $S\ H$ — or suction heating, the temperature rise of the charge in the manifold and the cylinder—for the several cases examined, will be constant. While this may not be strictly true, neither is it very far wrong. The assumption is made also that suction resistances and the like are the same for the several cases. This, obviously, is unlikely, since the resistance to flow of a complicated

vaporizer will be higher than that of a simple carburetor. This, as well as other assumptions which will appear from time to time, is made to establish limiting conditions. Likewise, it is assumed that exhaust-stack conditions are such that the pressure at the end of the exhaust stroke is equal to that of the atmosphere. We have, therefore, $P_A = T_{EX}\ P = 14.7$ lb. per sq. in. absolute in all cases and T_A = dry-mixture temperature = 510 deg. fahr. absolute for aviation gasoline.

B—Compression:

Examination of the literature indicates that several experimenters [10] have determined the value of the exponent for the compression stroke to be 1.33, approximately; that is $PV^{1.33}$ = a constant, for compression, in all cases. For simplicity, the mean effective pressures for both compression and expansion have been calculated on the basis that the piston displacement equals unity. Since the compression ratio = 5.42 (displacement + clearance/clearance) and the displacement = 1.00 we have: Clearance = 0.226.

The mean compression pressure is:

$$\frac{P_B V_B}{s-1}\left[R_v^{s-1}-1\right]$$

or

$$\frac{14.0 \times 1.226}{1.33-1}\left[5.42^{0.33}-1\right]$$

or 39.5 lb. per sq. in. absolute.

The compression pressure is

$$P_C = P_B \times R_v^{1.33} = 14.0 \times 5.42^{1.33}$$

= 133.0 lb. per sq. in. absolute. The compression temperature is

$$T_C = TST \times R_v^{s-1} = 552 \times 5.42^{0.33} = 965 \text{ deg. fahr. absolute.}$$

C—The Combustion Line and Maximum Pressure:

For constant volume combustion, we have:

$$Q = WC_v(T_2 - T_1) \text{ and } T_2 - T_1 = Q/WC_v\ ;$$

where Q is the heat added, W is the weight of the working gases and C_v is the mean specific heat of the products of combustion. The calculation can best be made by taking $W = 1$.

It is necessary to determine the amount of heat added to the charge. This is equal to the heat per pound of mixture modified by combustion efficiency E_f, by the heat loss due to dissociation S, by the heat loss due to radiation during combustion μ, and by the ratio of charge weight to total weight of the working gases, N_R.

For the combining proportions of air and fuel, combustion will be complete; hence, $E_f = 1.00$.

The heat loss[11] due to dissociation, S, is assumed to be 2 per cent; therefore, $1-S = 0.98$.

The heat loss[12] due to radiation, μ, *is assumed to be* 0.12; therefore $1-\mu = 0.88$.

The volumetric efficiency is known to be 0.906 at 1700 r.p.m. For gasoline rated at 20,000 B.t.u. per lb. and an air-fuel ratio of 15, we have 1250 B.t.u. per lb. of mixture. Therefore:

$$Q = 1250 \times E_f \times (1-S) \times (1-\mu) \times E_v$$
$$= 1250 \times 1.00 \times 0.98 \times 0.88 \times 0.906$$
$$= 978 \text{ B.t.u. per lb. of working gases.}$$

Calculation of the mean specific heat of the products of combustion is made by using the thermal-energy values of Goodenough and Felbeck[13]. For this case, $C_v = 0.3427$. Using the foregoing values, we have:

$$T_2 - T_1 = (987/0.3427) = 2850 \text{ deg. fahr.}$$
$$P_D/P_C = T_D/T_C \text{ and } P_D = (T_D/T_C)P_C$$
$$P_D = 133 \times [(2850 \times 965)/965]$$
$$= 526 \text{ lb. per sq. in. absolute.}$$

D—Mean Expansion Pressure:

The mean expansion pressure is:

$$\frac{P_D V_D}{S-1}\left[1 - \frac{1}{R_v^{s-1}}\right]$$

It is known that the value of the expansion exponent varies with operating conditions, both as to its instantaneous and its mean values. The work of Ricardo[14] indicates that the value of S varies inversely as the number of British thermal units per pound of working gases. His values are used in these calculations, being taken from the curve in Fig. 14.

For the case in hand $S = 1.17$, and we have, for the mean expansion pressure:

$$\frac{526}{0.17} \times 0.226\left(1 - \frac{1}{5.42^{0.17}}\right) = 174.5 \text{ lb. per sq. in. absolute.}$$

The pressure and temperature at the end of expansion are determined from the maximum pressure as used in the equations $R_P = R_v^s$ and $R_T = R_t^{s-1}$. Thus:

$$P_E = 526\ \frac{1}{5.42^{1.17}} = 72.8 \text{ lb. per sq. in. absolute.}$$

$$T_E = 3815\ \frac{1}{5.42^{0.17}} = 2860 \text{ deg. fahr. absolute.}$$

E — Gross Indicated Mean Effective Pressure.—Since this equals the mean expansion pressure less the mean compression pressure, we have 174.5 — 39.5 = 135 lb. per sq. in.

F — Brake Mean Effective Pressure.—This equals the gross indicated mean effective pressure less the losses in mean effective pressure. For the engine under discussion, the "losses mean effective pressure" is given in the test record[15] as 16 lb. per sq. in. Hence the brake mean effective pressure is 135 — 16 = 119 lb. per sq. in. The test report referred to gives 117 lb. per sq. in. at 1700 r.p.m. as the brake mean effective pressure.

While the method used in the foregoing calculations is simple and the correction factors may be somewhat modified in the event that more accurate data should become available, it will be seen to give results which agree rather well with test records—at least for the case in point. It is believed that the collection of many data from complete tests will make possible the determination of values for these correction factors within narrow limits, which should simplify greatly the prediction of performance.

I wish to express hereby my appreciation of the assistance of Dr. Charles E. Lucke in the preparation of this paper.

[10] See *The Automobile Engineer* (London), vol. 11, June, 1921, p. 219; Some Experiments on Supercharging a High-Speed Engine, by H. R. Ricardo.

[11] See The Internal-Combustion Engine, by D. R. Pye; Oxford University Press, London, 1931.

[12] See The Airplane Engine, by L. S. Marks; McGraw-Hill Book Co., 1922.

[13] See University of Illinois Bulletin No. 139; An Investigation of the Maximum Temperatures and Pressures Attainable in the Combustion of Gaseous and Liquid Fuels, by G. A. Goodenough and G. T. Felbeck.

[14] See *The Automobile Engineer* (London), vol. 11, June, 1921, p. 219; Some Experiments on Supercharging a High-Speed Engine, by H. R. Ricardo.

[15] See National Advisory Committee for Aeronautics Report No. 102; Performance of a Liberty-12 Airplane Engine, by Sparrow and White.

Noise Treatment in the Automobile

By Theodore M. Prudden
Pacific Mills

NOISE reduction in an automobile covers two fields—mechanical quiet and acoustical quiet. This paper deals with acoustical quiet, which is the quiet obtained by the application of scientific acoustics to the car body. It consists in shutting out extraneous noises and absorbing those unavoidably present in a car. It is distinct from the quiet achieved by precision workmanship.

Perhaps the most important point is that there are two divisions of adequate noise treatment. These two divisions are insulation against noise and the absorption of noise. Although they are quite different, they are sometimes confused. Both of these means must be used to obtain satisfactory results, for real sound treatment of a car consists in not merely shutting out incoming noises (that is sound insulation) but in absorbing these noises which arise inside the car, or which leak in. A car which is merely insulated against the penetration of outside noises is like a leaking boat which has no pump. Eventually the water will leak into this boat until the level of the water inside the boat is the same level as the water outside. Similarly a sound-insulated car cannot completely stop the noises from coming in, and how much they come in will depend on how many openings there are, such as open windows.

The noise level inside a car will build up until it is the same as the noise level outside, provided there is no absorption of whatever noise does come in. Sound-absorptive material is the pump which reduces the noise that does enter the car. Hence both sound insulation and sound absorption are essential to treat a car properly.

Absorption is more important than insulation because it is possible to quiet a car if enough absorption alone is used, but insulation alone can never do it, when the windows are open.

Another distinction between acoustical quiet and mechanical quiet is that the mechanical quiet will disappear gradually as the car grows older, due to mechanical wear, whereas acoustical quiet achieved with the proper materials continues to function with very nearly the same efficiency, no matter how old and noisy the machinery becomes.

The results from adequate use of acoustical quiet are not always what one would expect. Of course, there is primarily the reduction in noise, but after an hour's ride in a quieted car, the memory of its previous noisiness is often lost. A more tangible asset is that a quieted car can be driven more miles in a day without fatigue.

Another result from quieting the car is that you can drive such a car faster without realizing your increase in speed. This is probably because we are accustomed to judge our speeds unconsciously by the noise that our car is making. Reduce the car noise and you unconsciously run faster until the increased speed of the car reaches the noise level to which you are accustomed. This statement is readily demonstrable and is one of the positive proofs of noise reduction in a car.

One of the most valuable improvements from an acoustically treated car is that the sounds of the whole car are the sounds that one gets from a much higher-priced car. A $3000 car, with its heavier construction, has muffled noises from the wheels, and little or no engine noise, whereas in a less expensive car one expects to have a certain amount of engine noise and more intense road noises. The sound treatment gives it the noise characteristics of the heavier, expensive car.

The reason for this unexpected improvement is probably because all absorptive materials are very much more efficient in absorbing high-pitched noises than they are in taking up the lower-pitched ones, thereby leaving more of the low noises in the car, and since these lower-pitched ones are the noises arising from heavier construction, it obviously follows that a noise-treated car seems to have the characteristics of a more-expensive construction.

One of the greatest gains is the reduction of tire noises, particularly on rough roads. This is difficult to demonstrate around Detroit, because the roads here are so superlatively fine. If you drive over the rougher surface macadam which is gaining popularity in New England, or over the brick roads of Ohio, you will be struck with their greater noisiness. With an adequately noise-treated car, it is possible to drive from a smooth surface road on to a noisy one without being conscious of the difference of surface.

Vibration and Noise

There is an aspect of noise reduction about which little is known. It is the connection between sound and vibration. It is a well attested fact, often noted by earthquake authorities, that a moderate earthquake which is not heard is seldom felt. For instance, if an earthquake shakes a building in which there are two adjoining offices and one office has pictures on the walls or other furnishing which can rattle, the occupants of the room with the pictures will feel the earthquake because they hear it, and the occupants of the adjoining silent room will not even suspect that it has occurred. This phenomenon of joining two senses is similar to the phenomenon of a man being unable to taste the smoke of his pipe in the dark if he cannot see it. Noise reduction tends to reduce the feeling of vibration, even though the vibration itself has not been reduced.

We have found surprising variation, so far as the ear could tell, from putting the same amounts of acoustic material in different cars. For instance, the effect in a high-priced car is often less obvious than in a lower priced car. These differences are possibly due to the unreliability of our ears in judging noisiness. Measurement by scientific instruments may show a substantial noise reduction, but if the ear does not feel this reduction, the car builder is naturally doubtful of its value. Perhaps this is due to the fact that the human ear is affected not only by the difference in loudness between

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two noise levels, but also by the difference in the pitch of the noises. If two noises are of equal loudness, yet one noise is of higher pitch than the other, the higher pitched noise will be more objectional even though it is no louder. Conversely, the lower pitched noises may be louder than the high-pitched noises, yet be no more annoying. It is theoretically possible to have a noisy car with low-pitched noises, less objectionable than a quieter car with high-pitched noises.

Another weakness of our senses is the impossibility of retaining in one's memory the differences one has experienced in noise changes. Just as you cannot tell whether the room you enter is 10 deg. cooler or warmer than another room you were in an hour before, so it is with noise levels. You cannot recall the comparative noise level of a car you rode in an hour before—to say nothing of a day before.

There is time to touch on only a few of the details of treating a car. In the first place, the most economical use of acoustic material will be in such places where both sound absorption and sound insulation can be effected simultaneously. This means that the ideal acoustical material should have the two properties combined. That there have not been such materials available in the past is perhaps one reason why so little use has been made of acoustical quiet. The places where these two effects can be obtained are on the dash and shroud, on the roof of the car, and on the floor. In other places the installation of acoustic material utilizes only part of its properties. For instance, there is little advantage in placing sound-absorbing material behind the upholstery or on the doors behind the fibre panels, for in these places the noise inside the car cannot readily penetrate the upholstery or fibre panels to reach the sound-absorbing materials underneath. Hence in such places the material acts only as a sound insulator, and probably a less-expensive material can be used. Conversely, material placed on the back of the front seats, does not act to insulate against sound, but only as a sound absorber.

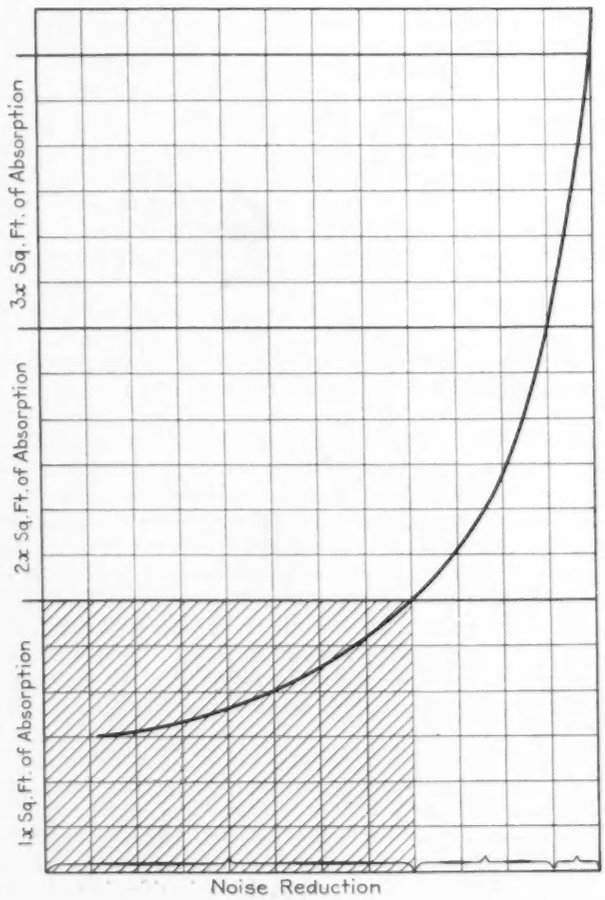

Fig. 1—Results Obtained from the Use of Typical Sound-Absorptive Material. The First Application of *X* Sq. Ft. of Material Accomplishes a Great Reduction in the Noise Level. Subsequent Applications of *X* Sq. Ft. of Material Produce Less and Less Effect.

It is common practice to install heat and sound insulation materials on the dash and toe boards, but since the areas for sound absorption are so limited, it is necessary to use also the whole floor for sound absorption, if adequate noise reduction is to be obtained. Hence material put on the floor should have high absorption as well as insulative value. To achieve high absorption in a floor material, it is essential to have the floor covering of such a nature that the noises within the body can penetrate through the floor covering and reach the absorptive material underneath. This means that carpets made with a rubber backing cannot be used to advantage, because the rubber backing will keep the sound from passing through to be absorbed. Similarly a rubber mat for the front floor boards is even more detrimental in stopping sound penetration. This need for an open carpet, both in front as well as in the rear, is contrary to the common practice in lower-priced cars, but it is a change which will have to be faced if this desirable floor space is to be used for absorption.

The shroud and dash are the noisest places in most cars. Many cars today already use materials of high insulative value on the dash to reduce the entrance of engine noises, but the rigidity of these materials prevents their use on the curve of the shroud. In addition to insulating, it is wise to face the inside of the dash with sound-absorbing material, taking care to seal up the holes in the dash where the wires, speedometer shaft, etc., pass through. The lining of the shroud with absorptive material is such common practice that little need be said about it, except that such lining material should be an insulator as well as absorber,—the metal of the shroud readily transmits sound. The kick pads on either side of the shroud are often made of a fibre board faced with fabric. With such an arrangement, the absorptive material underneath is covered by the fibre board and not readily penetrable by sound. Of course, these kick pads are usually open at their upper end so that sound can flow down in back of the fibre board, but more benefit from the sound absorptive material will be obtained if the center of this kick pad is cut out in the shape of a board frame, over which the cloth is stretched, so that the sound need only strike through the cloth to reach the absorptive material in back.

It would seem as though the space in the underside of the deck would be one of the most effective places to put acoustical material but that is not the case. Actual tests have shown that the cotton batting and burlap in the deck have of themselves considerable absorptive properties. Therefore, any material which is put on the deck will not be adding the percentage of absorption as though there were no absorptive material already there. In other words, if the cotton batting has already 15 per cent absorption and the material with which you plan to treat a car has 25 per cent absorption, you will only gain an addition of 10 per cent absorption over what was already in there. The curved metal edges of the roof are, however, ideal places for sound treatment. They furnish a considerable area (from 8 to 15 sq. ft.) and give to

the car the full value of the material in heat and sound insulation and absorption. Also absorptive material placed here is more nearly on the level of one's ear and its effect is more pronounced.

From these few details, it will be evident that no one material is suitable for all places in the car. It is obviously more economical to use a wall board at 2 cents per ft. under the rear seat where only sound insulation is necessary, than to put there a material which is both a sound insulator and an absorber.

There is not time to discuss the different acoustic materials which may be used in a car, so I am merely listing the properties which experience has shown to be essential:

High absorptive-efficiency
Sound-insulation efficiency
Thickness of material
Flexibility
Freedom from odor
Weight
Durability
Heat-insulative value
Ease of cutting
Cost

Cost has been placed last, not because it is not of great importance, but because an acoustic material must first fulfill the practical requirements before its cost is worth considering.

One of the points to be determined is how far to go in the installation of acoustic material. There has to be an economic limit somewhere.

Consider Fig. 1. Here you will see plotted the results obtained from a typical absorptive material. On one side are marked off equal units of x sq. ft. of such material; and on the other side are shown typical decibel reductions accomplished by the material. Notice that the first x sq. ft. of material accomplish a great reduction in the noise. Notice how the second x sq. ft. accomplish less reduction, and the reduction continues to be less for each additional unit.

Now, the first x sq. ft. might represent the absorptive materials which are already in an untreated car. Such materials are the upholstery, carpets, the cotton batting in the deck, etc.; hence the first big reduction has already been achieved, and only lesser reductions can follow. It will readily be seen that from a fourth or fifth unit of x sq. ft. the noise reduction would be so small that it would hardly be worth while. Between $2x$ and $3x$ sq. ft. is about the economic limit. This chart represents the reductions accomplished by a material having the absorption of 1½ in. of hair felt. A less-efficient absorber would, of course, require a larger number of square feet to achieve the same results.

Another point to be determined is whether to use 30 sq. ft. of a very efficient absorber at a higher cost, as compared with 60 sq. ft. of half as efficient material which will cost less.

In using the less efficient absorber, the engineer will find himself up against a problem of being unable to locate areas within the car where the material could be installed, or where it would function easily.

A practical example of this condition would be comparing ¾ in. jute felt, having an absorption of perhaps 30 per cent, with 1½ in. of jute felt whose efficiency might be 60 per cent. It is evident that 30 sq. ft. of the 1½ in. felt would be nearly as absorptive as the 60 sq. ft. of the ¾-in. jute. The 30 sq. ft. of the 1½-in. jute felt could be installed on the floor, shroud and dash, whereas it would be necessary to install the ¾-in. jute in the same places and also on the backs of front seats, sidewalls, and under the headlining, in order to get in

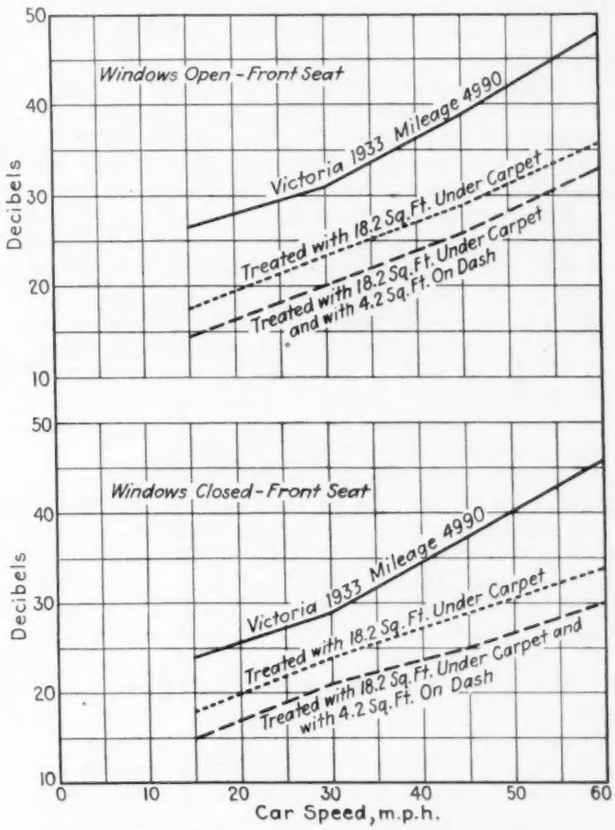
Windows Open - Front Seat
Victoria 1933 Mileage 4990
Treated with 18.2 Sq. Ft. Under Carpet
Treated with 18.2 Sq. Ft. Under Carpet and with 4.2 Sq. Ft. On Dash
Windows Closed - Front Seat
Victoria 1933 Mileage 4990
Treated with 18.2 Sq. Ft. Under Carpet
Treated with 18.2 Sq. Ft. Under Carpet and with 4.2 Sq. Ft. On Dash
Decibels
Car Speed, m.p.h.
0
10
20
30
40
50
60

Fig. 2—Front-Seat Noise Level at Different Car Speeds, with and Without Acoustical Treatment

the necessary 60 sq. ft. The disadvantages or difficulties in putting material in any of these extra places are obvious.

I should like to say a few words regarding the units used in the measurement of sound. Sound is measured in terms of a unit called a decibel. This term comes from the name of Alexander Graham Bell of telephone fame, and is one-tenth of a "bel" just as a "decimal" is one-tenth of a unit. A decibel may be roughly described as the smallest change in sound intensity which the human ear can appreciate. The quality of a sound insulator depends on its "decibel reduction" properties, or in other words how many of these decibel units are reduced below the level of hearing as the sound tries to pass through the material. Absorption is measured in terms of percentage of absorption. An open window gives 100 per cent absorption but it also gives zero decibels of noise reduction, because it does not resist passage of sound.

The two instruments commonly used in the automobile industry for measuring noise are the Western Electric 3-A Audiometer and the Burgess Acoustimeter. The Western Electric Audiometer is the simplest of the noise-measuring instruments. The reader controls the loudness of an electrically made buzz which is transmitted to a receiver on his ear. He tunes down this buzz until the sound is lost, which means that the noise of the buzzer is masked by the noise in the car. He then takes his reading directly from the dial on the instrument.

The Burgess Acoustimeter is direct reading and does not depend upon the hearing of the observer. It is a more bulky and more expensive instrument.

The biggest problem in noise tests is to reduce to a minimum all outside noises which can affect the noise levels in

the car. It is obvious that every car will seem to be quiet in very noisy traffic because the noises within the car are masked by the greater outside sounds.

The single greatest obstacle to noise testing is wind. We have determined that side or quartering wind of only 20 m.p.h. can raise the noise level in the car as much as 10 decibels. This raises the noise level more than we can usually lower it by adequate noise treatment. The most ideal conditions are with a wind less than 10 m.p.h.

The choice of road for testing is important. If possible, it should be of such direction that you drive directly into the wind. It also should be as nearly level as possible, so that there will be little variation in the power necessary to ascend grades. The road should be dry, and should be free from snow and snow banks along the edge, because snow is in itself a good absorber and reduces the noise levels.

Rain is bad, of course, because the drumming of the rain drops increases the noise level in the car.

When the road passes between hills or buildings, or through a cut, such obstructions cause the wind to sweep down the road in gusts and also tend to reflect back irregularly the sound from the car.

Other factors affecting the noises, which should be eliminated, are openings in the hood, or the dash ventilator, the various settings in ride control mechanisms, and the pressure of the tires. Also be sure to keep the same number of observers in the car between different tests. This is because the clothing of each person adds an appreciable amount of absorptive material.

A reduction in the noise level of 5 decibels is usually readily noticeable; in architecture it is an achievement, but it is possible through the use of materials which are both sound insulators and sound absorbers of sufficient efficiency, to obtain a reduction in the noise level in a car of from 8 to 10 decibels.

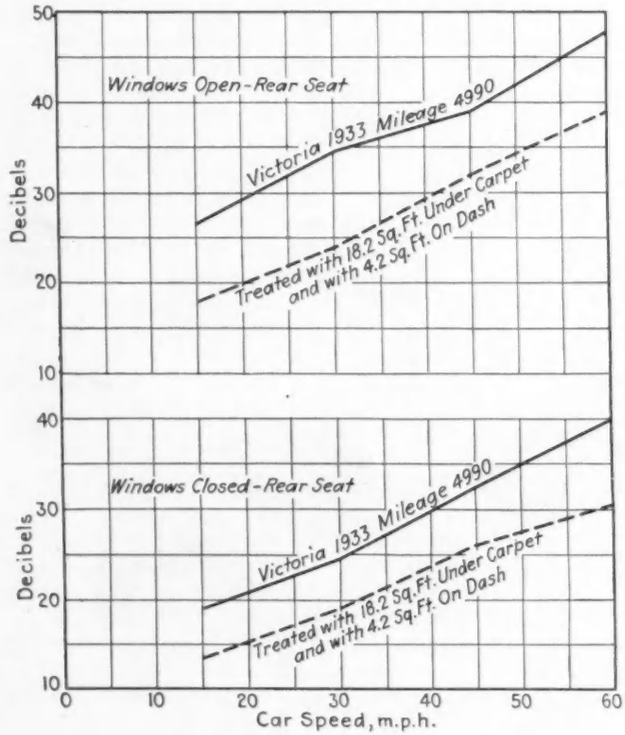

Fig. 3—Rear-Seat Noise Level at Different Car Speeds, with and Without Acoustical Treatment

How much improvement this achieves depends upon what was the noise level at the start. If we reduce the noise 10 decibels—i.e., from 40 decibels to 30 decibels—the improvement is not as noticeable as if we reduced it from 30 decibels to 20 decibels. In the first case we have reduced it 20 per cent; in the second place we have reduced it 33 per cent.

By adequately treating a dining car, the Pennsylvania Railroad reduced the noise level 10 decibels. This was at a noise level of 60 decibels. In other words, they accomplished a reduction of 16½ per cent, yet this reduction of only 16½ per cent was enough so that passengers entering the car would comment on its quietness and ask if the car had rubber mountings.

I am a little handicapped as regards having actual graphs showing noise reduction in a car, because most of such graphs have been obtained in conjunction with different automobile companies and are naturally confidential. I have, however, the independent results obtained in a 1933 model which I show as Figs. 2 and 3. The reductions obtained in this case are much greater than usual and cannot be considered an average case. The average noise reduction in this case and covering all conditions is 10.4 decibels.

The reasons why so remarkable reductions were accomplished here are probably due to the sheet metal floor of this car, which floor has in itself so little insulation that the acoustical material functioned to greater advantage than is usual.

Notice that in the front seat (Fig. 2) a separate test was made showing the value of the floor material without the dash covering, and then a further test after the dash material was applied. These extra tests were made to show how effective this particular dash mat was. Here is a case where the standard materials of the dash insulation are inadequate sound insulators.

Further proof of how the noise comes through the dash is shown by the greater noise in the front seats, as compared with the rear seats.

You will notice that in the front seat with the windows open, the noise level of the treated car at 45 m.p.h. is almost identical with the noise level of the untreated car at 15 m.p.h. or at 60 m.p.h. treated, is the same as 35 m.p.h. untreated.

In the front seat with the windows closed, even better conditions prevail.

The average increase of speed possible in the treated car without any increase of noise is 23 m.p.h. Notice also, in the rear seat, when the windows were open, that the treated car is slightly quieter than in the same seat with the car all closed up. In the front seat that the treated car, with the windows open, is very much quieter than the untreated with the windows closed. Having as much or greater quietness in the open car as in the car with the windows all closed, is perhaps one of the most remarkable evidences of the noise reduction obtained.

Summary

To sum up: It is necessary to apply both sound absorption as well as sound insulation to a car to adequately quiet it, but there are materials now available to accomplish worthwhile results; and the results improve the characteristics of the whole car, not merely the body.

I believe that the installation of acoustical quiet will eventually become standard equipment in the automobile world, as it has already become standard in other industries. Chassis design has progressed to such refinement that major improvements will be increasingly rare. It is evident that the evolution of the automobile will tend toward greater improvement in body design; i.e., comfort. This is the next step.